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Spray Cooling Enhancement of Air-Cooled Condensers

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REPORT SUMMARY

Dry cooling of power plants may be an attractive alternative to wet cooling, particularly where water conservation and environmental protection pose critical siting issues. However, dry cooling technology may be unable to maintain design plant output during the hottest periods of the year, which are often periods of peak system demand. This study—cosponsored by EPRI, the California Energy Commission, and Crockett Cogeneration Co.—evaluated the use of a low-pressure spray enhancement system to address the efficiency and capacity penalties of dry cooling. Such a spray enhancement system would offer a number of clear advantages in the form of decreased energy requirements, lower capital expenditures, and reduced operation and maintenance (O&M) costs.

Results & Findings

Inlet air spray cooling involves spraying a small amount of water into the inlet air stream of an air-cooled condenser (ACC) where it evaporates and cools the air. Incomplete evaporation introduces a number of challenges. Unevaporated droplets contribute nothing to the cooling effect but add to the plant water use if they cannot be collected and recycled. If they are carried into the ACC's finned tube bundles, the possibility for surface scaling and corrosion arises. If they fall out of the air stream as "rainback," they may constitute both a maintenance problem and an environmental discharge violation. While extremely fine sprays can facilitate complete evaporation, this technique results in a high-pressure, low-nozzle-flow design that proves very costly. The primary objective of the project, therefore, is to determine whether a reasonably priced spray enhancement system using moderate-pressure, higher flow nozzles with or without demisters can provide adequate cooling, high fractional evaporation, and acceptable "rainback" without endangering the integrity of the finned tube heat exchangers. Field testing revealed that

- The cooling effect was a strong function of ambient wet bulb depression and spray flow rate. The attendant cooling effect ranged from 60 to nearly 100 percent of the wet bulb depression, depending on spray rate and ambient conditions.
- At conditions where the use of spray enhancement would most likely be considered (ambient temperature > 90° F and relative humidity < 40 percent), a cooling effect of 80 percent or greater of the wet bulb depression could be expected.

Challenges & Objectives

Dry cooling suffers from the inability to maintain design plant output during the hottest periods of the year. Depending on meteorology at the site and the choice of design point, a plant can experience capacity reductions of 10 to 20 percent on the steam side alone because of increased turbine back pressure. The problem is compounded since these hours represent peak load periods for most power systems and, for combined cycle plants, the gas-turbine side suffers simultaneous

capacity reductions from increased ambient temperatures as well. This study addresses the question of whether an inlet spray cooling system can significantly enhance performance of an ACC at a dry-cooled power plant, while maintaining acceptable water use and reasonable cost, minimizing energy requirements and complexity, and avoiding unacceptable O&M requirements or environmental impacts.

Applications, Values & Use

While additional evaluation is required, test results and qualitative observations suggest that an economical and reliable approach can be developed for spray enhancement of air-cooled systems. Estimates have been made on the cost of a 30-cell ACC intended to approximate the size typically found at the type of 500-MW combined-cycle plant proposed for development in California. Such an ACC—with performance capability to reduce the inlet air temperature by 30 degrees F at the hottest conditions—was estimated at $600,000. A recent study of dry cooling system costs estimated that the application of such a system at a hot, arid site (based on meteorological conditions similar to Blythe, California) would recover 75 percent of the output loss during the hottest 1000 hours of the year. The recovered output was valued at $250,000 to $1 million, depending on the assumed price of power during peak demand periods. This analysis suggests a payback time ranging from less than one year to two years.

EPRI Perspective

Advantages of a dry cooling system include modest water use, appealing initial cost, and the ability to retrofit existing units. Disadvantages include the increased potential for scaling and corrosion of air-cooled heat exchanger surfaces, particularly in areas where high ambient temperatures exist. One approach to alleviating this condition is the use of inlet spray cooling—with its attendant challenges of unevaporated spray—to enhance dry system performance during brief hot periods and restore much of the lost plant capacity. Further EPRI work to minimize or eliminate unevaporated spray will focus on nozzle array optimization, advanced spray devices, water purification and management, and droplet capture and return. EPRI believes that work in this area will ultimately result in viable spray enhancement technology for power plant use during peak system demand periods.

Approach

Investigators screened several spray nozzles at the EnviroCare laboratory in Napa, California, and selected promising designs for field testing on a single cell of a Balcke-Duerr air-cooled exchanger at the Crockett, California Cogeneration plant—a 220 MWe gas-fired, combined-cycle plant. In addition, they tested three mist eliminator designs to determine whether they could be used to prevent impingement of unevaporated droplets on the finned tube surfaces. Single-cell tests helped determine the effect of spray droplet size, ambient meteorology, spray flow rate, and nozzle location on inlet cooling and efficiency of water use.

Keywords

Air-cooled condensers
Spray cooling
Dry cooling

ABSTRACT

Dry cooling of power plants may be an attractive alternative to wet cooling where water conservation and environmental protection are critical siting issues. However, the technology may be unable to maintain design plant output during the hottest periods of the year, which are often the periods of peak system demand. Capacity shortfalls because of increased turbine backpressure, if sufficiently large and widespread, could create both a potential system reliability problem and a substantial revenue loss to the plant owners.

This work, supported by the California Energy Commission, EPRI, and Crockett Cogeneration Co. investigates inlet spray cooling as a potential solution to this problem. A few degrees reduction in the inlet air temperature can restore much of the lost plant capacity during hot hours. Advantages of this system are modest water use, modest initial cost, and the ability to retrofit existing units. Disadvantages include the increased potential for scaling and corrosion of the air-cooled heat exchanger surfaces.

This paper presents preliminary results from lab tests conducted at the EnviroCare laboratory in Napa, California and from field tests performed on a single cell of a Balcke-Duerr air-cooled exchanger at the Crockett Cogeneration plant—a 220 MWe gas-fired, combined-cycle plant in Crockett, California. The objectives of the program were to identify and evaluate the use of a low-pressure spray enhancement system to address the efficiency and capacity penalties of dry cooling. Low-pressure systems would have the advantage of lower energy requirements as well as lower capital costs and less maintenance, which would lead to lower operation and maintenance (O&M) costs. One difficulty of low-pressure systems that needs to be addressed is the development of nozzles that provide a uniformly small droplet size (for improved evaporation and control of deposition on the ACC), while maximizing flow rate and minimizing energy requirements and complexity.

The performance of several spray nozzles was screened in the laboratory and three were selected for field tests. Single-cell tests were carried out to determine the effect of spray droplet size, ambient meteorology, spray flow rate, and nozzle location on the inlet cooling effect and the efficiency of water utilization. In addition, observations of system operating behavior relevant to the design and O&M of a full-scale unit were made.

Test results and qualitative observations suggest that, while additional testing is required, an economical and reliable approach can be developed for spray enhancement of air-cooled systems.

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EXECUTIVE SUMMARY

California has experienced a growing need for both electric power and water for decades. Meeting this need can lead to conflict between water for power plant cooling requirements and water for agricultural, residential, industrial, and environmental uses. Therefore, many consider the use of dry cooling to be an attractive alternative to the use of wet cooling on the grounds of water conservation and environmental protection. It is well known that dry cooling, while substantially reducing the need for water in power plants, can cause reductions in plant efficiency and output during the hotter periods of the year.

One approach to alleviating this condition is the use of some water to supplement or enhance dry system performance for these brief hot periods. This study analyzed and tested the use of inlet air spray cooling, in which a small amount of water is sprayed into the inlet air stream of an air-cooled condenser where it evaporates and cools the air.

Questions to be Answered

The study addresses the question of whether an inlet spray cooling system can significantly enhance the performance of an air-cooled condenser at a power plant while maintaining an acceptable water use and reasonable cost without incurring unacceptable O&M requirements or environmental impacts.

The degree to which the water sprayed into the inlet air stream evaporates raises a number of the important questions. Incomplete evaporation introduces several problems. Unevaporated droplets contribute nothing to the cooling effect but add to the plant water use if they cannot be collected and recycled. If they are carried into the ACC's finned tube bundles, there is danger of scaling and corrosion of the surfaces. If they fall out of the air stream as "rainback," they may constitute both a maintenance problem and an environmental discharge violation. Producing extremely fine sprays can approach complete evaporation, but results in a high-pressure, low nozzle flow design that is very costly.

Therefore, the primary objective of the project is to determine whether a moderate cost system using moderate pressure, higher flow nozzles with or without demisters can provide adequate cooling, high fractional evaporation, and acceptable "rainback" without endangering the integrity of the finned tube heat exchangers. This is approached in a combined laboratory and single-cell field test program with a cost benefit analysis based on the test results.

Laboratory Tests

Testing took place at EnviroCare, International's test facility on a 1/50th scale model of the full-size cell on an air-cooled condenser. Researchers tested several nozzles of differing design to select the most promising candidates for single-cell tests at full scale. They also tested three mist eliminator designs to determine whether they could be used to prevent impingement of unevaporated droplets on the finned tube surfaces.

Testing did not uncover a clear distinction in performance between different nozzle designs. Individual nozzles of both the internal swirl type and the impingement pin (or pintle) type were capable of achieving cooling effects of over 70% of the theoretical maximum cooling effect given by the wet bulb depression (defined as the ambient dry bulb minus the ambient wet bulb temperatures). The single-cell tests used two pintle-type nozzles (PJ-28 and PJ-20) and one swirl-type (WDB14-90).

Mist eliminators showed excellent performance in eliminating droplet carryover at face velocities up to and above those to be expected on operating ACC's. Full-scale testing used the Brentwood CF1900 film pack.

Single-Cell Tests

Researchers conducted field tests on a single cell of a full-scale air-cooled heat exchanger at the Crockett Cogeneration Plant located in Crockett, California. The site is located on the south shore of the Carquinez Straits, approximately 25 miles northeast of San Francisco. The plant, a 240 MW gas-fired, combined-cycle plant, is equipped with a 12-cell air-cooled steam condenser and a 3-cell air-cooled heat exchanger for plant auxiliary cooling loads.

Researchers conducted the tests by spraying water into the inlet air of the middle cell of the ACE, upstream of the fan. Spray nozzles were mounted on a rack, which could be raised or lowered to varying heights beneath the fan. Fifty to one hundred nozzles could be mounted on the rack in a rectangular array fitting within the perimeter of the fan shroud. Spray rates of 2 to 24 gpm could be provided at a 300 psi supply pressure.

The researchers recorded complete measurements of the flow rates and temperature distributions of both the hot side (a water/glycol mixture from the plant auxiliary cooling system) and the air side along with the spray flow rate. They determined the performance of the cell with and without spray enhancement for a wide range of operating conditions.

The field tests yielded the following general conclusions:

- The cooling effect ($T_{amb} - T_{inlet}$) was a strong function of ambient wet bulb depression ($T_{amb\ dry\ bulb} - T_{amb\ wet\ bulb}$) and spray flow rate.
- The effect of spray droplet size distribution and nozzle location (droplet residence time) was discernible but typically minor.
- The attendant cooling effect ranged from 60% to nearly 100% of the prevailing wet bulb depression depending on spray rate and ambient conditions.

- At conditions where the use of spray enhancement would most likely be considered (T_{amb} > 90° F and relative humidity < 40%), a cooling effect of 80% or greater of the wet bulb depression could be expected.

Cost-Benefit Analysis

A methodology was developed for the design and cost estimation of a full-scale inlet air-cooling system for an air-cooled condenser. The cost of a 30-cell ACC, intended to approximate the size typically found at a 500 MW combined-cycle plant of the type currently proposed for development in California, with performance capability of reducing the inlet air temperature by 30° F at the hottest conditions, was estimated at approximately $600,000.

The application of such a system at a hot, arid site represented by the "desert site" in a recent study of dry cooling system costs using meteorological conditions based on Blythe, California was estimated to recover 75% of the output loss during the hottest 1000 hours of the year. This recovered output was valued at $250,000 to $1,000,000 depending on the assumed price of power during peak demand periods. This analysis suggests a payback time ranging from 2 ½ to less than one year.

Recommendations

The primary remaining issue for the successful application of spray enhancement on ACC's is that of minimizing or eliminating unevaporated spray. Further work to address this question remains in four areas:

- Nozzle array optimization: Finding the best arrangement of nozzles to introduce the spray into the inlet air stream is important to minimize rainback, avoid site specific wind problems, and ensure a reasonably uniform distribution of the spray among the many cells. A field test at a full-scale operating plant in conjunction with a modeling effort could result in valuable design guidelines.
- Advanced spray devices: While reasonable cooling effects were obtained with conventional nozzles used in these tests, innovative techniques for producing finer mists at acceptable power consumption rates would be a significant advantage for the spray enhancement technology. Some industrial nozzle vendors claim recent advances which should be pursued. Additionally, some innovative concepts such as the use of nebulizers of the type developed for medical applications or adaptations of some rotating devices developed for agricultural (crop dusting) may have promise.
- Water purification and management: To the extent that it may be necessary to provide high-purity water for spray enhancement, the cost of treating the water and disposing of the resultant brine is an important design issue that could significantly affect the cost. An engineering study to develop economic design guidelines should be conducted.
- Droplet capture and return: A brief test was conducted as part of this study to evaluate the use of mist eliminators or cooling tower fill pack to ensure that no unevaporated droplets reach the finned surfaces. This may prove a more economical solution than the use of high-performance, fine mist nozzles or the production of very high-purity water. The test results

from this study were few and taken under conditions where evaluation of the effect of mist eliminators on performance was inconclusive, which warrants further tests.

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1
INTRODUCTION

This report documents the objectives, methods, and results of a study conducted for the California Energy Commission (CEC) and the Electric Power Research Institute (EPRI) to evaluate the effectiveness of using inlet sprays to enhance the performance of air-cooled condensers. Increased attention to the use of dry cooling at power plants worldwide, nationally, and in California—where the equitable allocation of the State's water resources is a crucial concern—has recently stimulated interest in this technology.

The need for power plant cooling water can come in conflict with agricultural, residential, industrial, and environmental requirements. Therefore, many consider the use of dry cooling to be an attractive alternative to the use of wet cooling on grounds of water conservation and environmental protection. However, dry cooling systems are more costly than comparable wet systems and their use can reduce plant efficiency and limit plant output during the hottest hours of the year. This is particularly important in California where the hottest summertime hours are those when power is most needed by the California grid. Capacity shortfalls of several megawatts because of increased turbine backpressure could create both a potential system reliability problem and a substantial revenue loss to plant owners.

In order to understand the consequences of a choice between wet and dry cooling, the CEC and EPRI recently sponsored a study to define, document, and explain the performance, economic and environmental trade-offs involved in the choice between alternative cooling systems. This report will briefly review the recently published results of that study[1] to establish the motivation for evaluating the spray enhancement technology.

Alternative Cooling System Trade-offs

Dry cooling suffers from the inability to maintain design plant output during the hottest periods of the year. Depending on the meteorology at the site and the choice of design point, a plant can experience capacity reductions of up to 10 to 20% on the steam side alone because of increased turbine backpressure. The problem is compounded since these hours are the peak load periods for most power systems and, for combined cycle plants, the gas-turbine side suffers simultaneous capacity reductions from increased ambient temperatures as well.

Comparisons of cost and performance for dry and wet systems were developed using a case study approach for four sites chosen to be representative of the range of climates encountered in California. For each site, an optimum wet and an optimum dry system were selected. The optimization methodology was designed to find the correct balance between initial capital costs, which increase as the size and cooling capacity of the cooling system increase, and the cost penalties incurred as a result of reduced plant efficiency and limited plant output, which increase as the size and capacity of the cooling system decrease. This relationship is illustrated schematically in Figure 1-1.

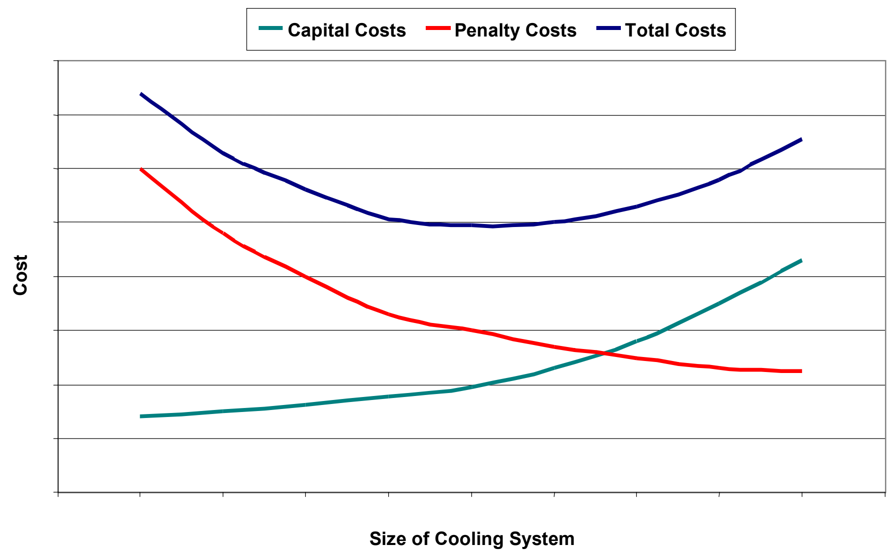

Figure 1-1
Schematic of Optimizing Considerations

The problem faced by dry cooling systems during periods of hot weather is best illustrated by the "desert" case. Figure 1-2 presents the temperature duration curve for this site and shows that an ambient temperature of 90° F is exceeded for approximately 1000 hours per year and 100° F for over 500 hours. Even for an optimized dry cooling system, the lost power output from the combined effects of efficiency reduction and capacity limitation during the hot hours exceeds 20,000 MWh per year. At a price of $50/MWh, this amounts to $1,000,000. If the price rises during the hot, peak load periods, as it has in the past to $500/MWh or higher, the loss exceeds $10 million per year. For comparison, the capital cost of the dry cooling system for this application was approximately $30 million.

By contrast, the costs for a dry cooling system in a location where these high ambient temperatures rarely or never occur are significantly less. The comparable design and optimization procedure applied to the "Bay Area" case, for which Figure 1-2 shows the temperature duration curve , results in a tower cost of $20 million and negligible penalty costs.

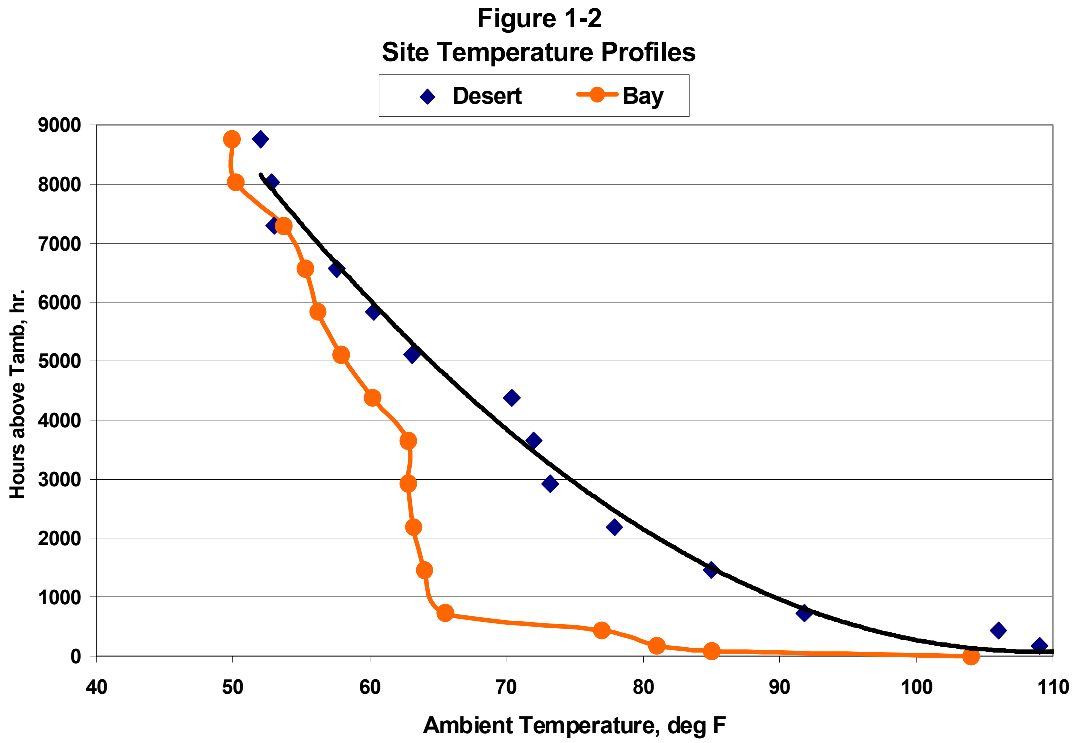

Figure 1-2
Site Temperature Profiles

One approach to alleviate this condition is the use of some water to supplement or enhance dry system performance for these brief hot periods. A number of approaches have been considered to accomplish this. They are reviewed in Section 2 of this report. One approach—inlet air spray cooling, which sprays a small amount of water into the inlet air stream where it evaporates and cools the air—was selected for detailed analysis and testing in this study.

Scope of Project

The objective of the study was to select and test an enhancement technology for air-cooled power plant steam condensers based on the use of a limited amount of water to reduce inlet air temperature during the highest temperature periods of the year. Low-pressure systems would have the advantage lower energy requirements as well as lower capital costs and less maintenance, which leads to lower O&M costs. One of the difficulties of low-pressure systems that needs to be addressed is the development of nozzles that provide uniformly small droplet size (for improved evaporation and control of deposition on the ACC), while maximizing flow rate and minimizing energy requirements and complexity.

The objectives were approached in four tasks:

- A brief review of alternate approaches to inlet air cooling and selection of a preferred candidate system for testing

- Preliminary laboratory testing to prescreen and select candidate components for larger scale field testing
- Field testing on an air-cooled heat exchanger at an operating power plant
- Evaluation of the expected costs and benefits of the application of the system to a full-scale unit

Organization of Report

The remaining sections of this report describe the methodology, results, and recommendations of the work conducted for the CEC and EPRI under this project, "Spray Enhancement of Dry Cooling."

Section 2 reviews alternative methods for achieving inlet air cooling on air-cooled condensers and the reasons for the selection of inlet sprays as the preferred system for testing. Section 3 reviews the state-of-the-art spray injection technology for air stream cooling and identifies similar systems used in other applications. Section 3 also discusses the issues to be addressed in the application of spray cooling to air-cooled condensers and defines desired characteristics for the application.

Sections 4 and 5 present the test portions of the work. Section 4 describes the laboratory tests and the application of the results to the selection of the system components and configurations tested on a full-scale air-cooled heat exchanger at the Crockett Cogeneration Plant. Section 5 describes the spray set-up and data acquisition system installed at Crockett and presents the results of preliminary system characterization tests done before the spray tests.

Section 6 describes the various elements of the test plan and presents, correlates, and interprets the data. Section 6 also discusses the operating experience and qualitative observations made during the course of the test program and interprets them in terms of the issues raised for full-scale application of the technology.

Section 7 uses the single-cell test data and correlations to carry out an analysis of the expected costs and benefits of applying the technology to a full-scale plant. Finally, Section 8 summarizes the conclusions of the work and presents recommendations for future development activities.

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2

ALTERNATE APPROACHES TO ACC ENHANCEMENT

There are a variety of ways to use a modest amount of water to enhance the performance of air-cooled condensers for those limited periods of the year during which the ambient temperature and the power demand are simultaneously high.

Hybrid (Wet/Dry) Systems

Conventional approaches use hybrid or wet/dry systems, where the heat is rejected through two separate cooling systems—a dry system which carries most or all of the cooling load during most of the year with the wet system picking up a portion of the load during the hotter periods when the performance of the dry system is limited.

These hybrid systems come in many different arrangements and designs. The recent EPRI/CEC study includes a brief review[8], and a 1993 CTI paper contains an excellent detailed description of the many system types[6]. While these systems can achieve significant water conservation and still maintain good hot-day performance, they can have high initial costs as a result of the need for two cooling towers (or a more complex integrated single structure), parallel circulating water loop components, more complex controls, and other requirements associated with providing two, nearly independent cooling systems. With the exception of plume abatement (rather than water conservation) designs, they have not been widely used in the past although there has been some renewed interest in recent years.

A second approach that can provide some degree of water conservation is the wet-surface evaporative condenser. This technology, which has been widely used in the chemical process and HVAC industries, has recently been considered for power plant use. A unit has been in use at MassPower—a 240 MWe plant near Springfield, Massachusetts—since 1993. Detailed descriptions of the technology are found in the recent EPRI/CEC study[8], in the CTI literature[12], on vendor Websites[13], and in vendor brochures.

Deluge Cooling

Another approach to performance enhancement is used in conjunction with all-dry, direct systems in which all the steam is condensed in an air-cooled condenser. The hot-day performance is enhanced by the use of water either to increase the heat transfer rates from the finned tube bundles (known as deluge cooling) or to precool the inlet air.

In deluge cooling systems, water is introduced onto the finned side of the air-cooled condenser tubes. In this arrangement, the tubes are horizontal and the fin surfaces are vertical. The water runs down the fins in a film with the air moving in cross-flow across the outer surface of the film. It transfers heat from the fin surface to the water film, conducting it through the film and rejecting it to the atmosphere through evaporation of the water at the outer surface of the film. It derives enhanced performance from both the higher heat transfer coefficient at the fin surface (compared to a dry fin rejecting heat to flowing air) and the temperature of the water in the film being lower than the ambient air. A version of this method is available commercially as part of the GEA-EGI Heller system[1;2]. It has also been studied analytically and experimentally as part of the EPRI/DOE Advanced Concept Test program[4].

Inlet Air-Cooling

In inlet air precooling systems, water is introduced into the inlet air stream of the air-cooled condenser. The water evaporates, reducing the air temperature to the finned tube bundles. The greatest temperature reduction theoretically achievable with this method is equal to the wet bulb depression, defined as the ambient temperature minus the ambient wet bulb temperature ($T_{amb} - T_{amb\ wb}$). In practice, only some fraction of that amount of cooling will be realized. As long ago as 1991, Conradie and Kroeger[3] showed that substantial increases in peak capacity on hot days could be achieved with modest (up to 70% relative humidity) inlet humidification/cooling. The systems can be designed in different ways.

Inlet Matrix or Packing

In these designs, the opening around the perimeter of the air-cooled condenser is fitted with panels of a mesh or matrix material through which the air can pass with minimum resistance and which can hold water in contact with the air stream. In some designs, the water enters at the upper edge of the panel and trickles down through the material while the air flows through the panel in cross-flow with the water. The system collects any unevaporated water at the bottom of the panel and recirculates it. Alternatively, the system can spray water on the upstream face of the panel and retain it in the matrix as the air flows past it.

Systems of this type have been used for inlet air-cooling on gas turbines and in agricultural applications, such as maintaining cool conditions in poultry sheds.[7] Mammoth Lakes Geothermal station tested such one on an air-cooled condenser and a planned project will install a full-scale system on another unit at that location.[11]

Inlet Spray Cooling

This approach involves the spraying of a small amount of water into the inlet air stream where it evaporates and cools the air. Gas turbine inlet cooling[9;10] and some process air-cooling applications have used spray systems. A system of this type has been operating at the Chinese Station Power Plant (a 25 MW waste wood-fired plant near Sonora, California) for several years and a recent test has been carried out on the air-cooled condenser at the El Dorado Energy Center's 500 MW gas-fired combined cycle plant near Las Vegas.

The National Renewable Energy Laboratory was the site for a recent comparative economic analysis of four of these systems as they would be applied to a small geothermal plant [5]. The study involved a deluge system, an inlet spray system, and two versions of an inlet matrix system. While the final analysis of the results is still underway, the initial results show a slight economic advantage to the spray inlet system with the highest return on investment of the four systems studied.

The inlet spray system was chosen for study in this project because of its relative simplicity, low initial cost, ease of retrofit to existing units, and the need for additional information on the effect of system design parameters and operating characteristics.

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3
INLET SPRAYS—SYSTEM DESIGN CRITERIA AND ISSUES

Spray Systems—An Order of Magnitude Analysis

A short analysis determined the additional energy output that might be obtained from a 550 MWe combined-cycle plant in California's Sacramento Valley as a function of water use. For climate conditions typical of the area, there are about 500 hours per year during which temperatures would be expected to exceed 80° F with simultaneous relative humidity below 50%. Spray enhancement used during those hours to humidify the inlet air to 85% would create inlet cooling of 15 to 25° F and increase plant output by approximately 4kW/gpm. A wet cooling system for a plant of this size would consume from 10 to 12 gpm per MWe from the steam portion of the plant or in the range of a billion gallons per year. Using 5% of that amount of water for spray enhancement would yield an additional 3,500 MWh per year with a value, depending on the assumed price of energy during peak periods, of perhaps $100,000 to $500,000.

Issues to be Addressed

The general question this study addresses is whether an inlet spray cooling system can achieve the benefits estimated above at acceptable water use and reasonable cost without incurring unacceptable O&M requirements or environmental impacts.

Water Consumption/Environmental Impacts

The amount of water that must be evaporated to obtain a given amount of cooling is a straightforward calculation. However, it is not practically possible to evaporate all of the water sprayed into the air under most conditions. Unevaporated droplets will collide with structural members, fan blades, fan shrouds, or other obstructions. They then coalesce into films and drain off as large droplets which fall onto the floor or ground below the unit as "rainback."

In some locations, this may constitute an environmental discharge violation. In any case, the water is lost to the cooling process. While it is possible to collect and recycle the water, it will probably require clean up with attendant cost, water consumption, and wastewater discharge. Therefore, the system design should provide the maximum fractional evaporation of the spray, leading to the general requirement of fine sprays and long residence times.

O&M considerations

The most important operating issue is the possible corrosion damage to, or scaling of, the finned tube surfaces through repeated wetting and drying, since it is virtually impossible to prevent all liquid droplets from reaching the heat exchanger inlet plane. There are two approaches to mitigate the problem. First, the use of high-purity, deionized water for the spray will eliminate scaling and corrosion even in the event of some surface wetting. This introduces water treatment costs and a wastewater discharge stream to address.

A second approach is the use of demisters between the fans and the heat exchangers. While keeping the finned surfaces dry, this method may both reduce the cooling effect by removing some of the evaporating droplets from the air flow prematurely and increase the rainback problem as it drains. Demisters on the inlet of the fans for each cell also represent an additional capital cost and may adversely affect fan performance.

Cost

The selected approaches for dealing with the water use and O&M issues discussed above are major determinants of system cost. The production of very fine droplets has led, in other applications such as gas turbine inlet cooling[1], to the choice of very high-pressure (~2000 psi or above) systems feeding a large number of small, low-flow (~ .05 to 0.1 gpm) nozzles. The system, which typically requires small diameter, stainless steel manifold piping and a number of small, high-pressure pumps, is prohibitively expensive for use on ACC's, which have a greater airflow per MW and a lower increase in power output for a given rate of water injection. In addition, as noted above, the use of either deionized water or demisters as alternate approaches to protecting the finned tube surfaces represents an important part of the system cost.

Therefore, the primary objective of the test program is to determine whether a moderate cost system using moderate pressure (<300 psi) and higher flow (0.2 to 0.5 gpm) nozzles with or without demisters, can provide adequate cooling, high fractional evaporation, and acceptable rainback without endangering the integrity of the finned tube heat exchangers.

Reference List

1. Inlet Air Spray Cooler for Gas Turbine Power Augmentation, EPRI, Palo Alto, CA: 1997. TR-108057.

4
LABORATORY TEST PROGRAM

Introduction

The choice of type and location of the nozzles through which the spray is injected into the inlet-air stream is important to the performance and cost of the system. Desirable characteristics are low pressure, high flow and small droplet size, since low pressure and high flow per nozzle reduce the complexity and cost of the delivery system while small droplets evaporate more quickly and cool more effectively. However, fine atomization typically requires higher pressures and smaller (lower flow) nozzles.

Droplets are typically formed by the creation inside the nozzle flow passages of sheets or ligaments that then break up through a combination of shearing action and surface tension. A number of general design categories exist:

- Internal swirl nozzles, in which flow passages inside the nozzle provide the shearing action to create the sheets and ligaments
- “Pigtail nozzles” in which an external corkscrew shaped element divides and shears off portions of the exit stream and imparts some swirl to the flow
- Pintle nozzles in which the exit steam impacts an impingement plate suspended on a hook-like support in front of the nozzle exit
- Colliding jet nozzles in which the flow is divided into two streams, which are then directed “head on” into one another in a variation of the pintle or impact approach

Modifications of these designs can provide differing attributes to the spray:

- A variety of cone angles from highly directed sprays with considerable forward momentum to nearly radial (pancake) sprays to provide broad coverage of an air stream
- Solid vs. hollow cone sprays—hollow cone is likely preferred for cooling applications since it provides better access of the air to the droplets

Project researchers conducted laboratory tests to aid in the selection of nozzles and mist eliminators and to provide general guidance for the arrangement of the inlet spray equipment to be used in the field-testing at the Crockett Cogeneration Plant. EnviroCare International, Inc. designed and built a pilot facility to simulate, at approximately 1/50th scale, a full-size cell on an air-cooled condenser. The 12-cell air-cooled condenser with 3-cell air-cooled heat exchanger at the Crockett Cogeneration Station served as the basis for the model/prototype approach.

Specifically, researchers developed an airflow rate, sprayed water rates, and spray distances to make performance extrapolations to full scale easier.

Objectives

The test plan included the following objectives:

- Evaluate the effectiveness of spray nozzle/mist eliminator systems for cooling of entering ambient air
- Compare the performance of candidate nozzles for achieving economic atomization of water droplets
- Fine tune, as appropriate, the water injection and interception strategies for consideration in full-scale field tests

Approach

EnviroCare designed and assembled the test system as shown in Figure 4-1. Testers determined velocity profiles and total airflow via pitot traverse, both in the inlet cylinder and downstream of the drift eliminator plane. Based on full-scale operating conditions and limitations, they decided that one airflow rate might be used for the testing because there was no perceived benefit to test at a condition that simulates half-speed operation of the fans at Crockett. Testing did include higher-than-design velocities for mist eliminators, because the testers decided that the mist eliminators could experience such conditions depending upon the full-scale installation approach and limitations.

Figure 4-1
Pilot Test Facility

Nozzle Tests

Researchers performed nozzle tests on 15 candidate EnviroCare-supplied nozzles and 1 "colliding jet" nozzle supplied through the Electric Power Research Institute. Visual testing of the nozzle spray patterns occurred at multiple supply pressures from 60 psig to approximately 340 psig. A ball valve located between the two supply pumps and test nozzle controlled the pressure and flow rate at the nozzles. Researchers evaluated spray patterns based on visual observations of the nozzle spray cone and the trajectory of the droplets once they left the nozzles They developed water flow rate and pressure relationships for a number of nozzles. Appendix A contains these data.

Researchers evaluated droplet size distribution primarily in a qualitative sense by witnessing spray droplet patterns and the apparent settling velocity of the droplets generated by the nozzles. They made additional characterizations by running their hands through the spray pattern itself. Even those less experienced with spray nozzle patterns could quickly discern the differences in apparent nozzle performance.

The EnviroCare-supplied nozzles exhibited a variety of patterns, depending on nozzle design. The nozzles with the impingement pintles (an impact disk positioned in front of the nozzle opening on the end of a "hook-like" support) were sensitive to the position of the hook/pintle and may be a maintenance problem in full-scale installations. These nozzles also typically exhibit a void, albeit less than 10–15 degrees of the full cone, in the spray pattern where the pintle base blocks the spray pattern. Additionally, the pintle generated some larger droplets, which fell out from the rest of the droplet population and would contribute to rainback. Figure 4-2 shows this.

Figure 4-2
Expanded View of Pintle-Type Nozzle Spray Pattern

Other nozzles with "swirl devices" exhibited equally good spray patterns, but in some cases had lower design flow rates (that is, <0.25 gpm) than might be desired. Table 4-1 summarizes the nozzles tested.

Table 4-1
Summary of Nozzles Pilot Tests

Test Date	Nozzles Tested	Style (1)	Approx. Flow Rate (gpm) @300psig
7/16/01	PJ -28	**P**	0.26
	PJ-32	**P**	0.42
7/17/01	24-6	**S**	0.45
	24-8	**S**	0.45
	JJS-5	**Custom Swirl**	0.22

7/18/01	24-20	**S**	
	24-18	**S**	0.15
	24-8	**S**	0.45
7/25/01	24-3	**S**	0.45
	20-8	**S**	0.35
	PJ-15	**P**	0.3
	24-18	**S**	0.15
8/7/01	PJ-28	**P**	0.26
	Gianotti	**Colliding Jet**	0.65
	WDB-24	**S**	0.52
8/8/01	PJ-28	**P**	0.26
	WD-24	**S**	0.52
	WD-8	**P**	0.06
10/22/01	PJ-24	**P**	0.22
	WDB-24	**S**	0.52
	SSD	**Custom Swirl**	0.34

(1) **P** Pintle-Type
S Swirl-Type

The colliding jet nozzle exhibited a spray pattern that was flat (that is, a cone angle of 180 degrees under conditions of limited transverse velocities), and appeared to have a population of droplets that were significantly larger (that is, 100–200 microns) than other candidate nozzles. This nozzle technology may have future promise. However, the researchers decided to continue the testing with nozzles that currently exhibit better spray patterns.

Photographs document visual perceptions of the sprays. Figure 4-3 is another example of the spray pattern from a candidate nozzle. The steel framework around the spray is the supporting structure, which houses the mist eliminators and induced draft fan. The wooden pallet in the background was used to reduce wind effects on the spray.

Figure 4-3
Example Spray in Pilot Test Facility

Nozzle/Mist Eliminator Testing

The evaluation of nozzle/mist eliminator testing took place in a custom duct using a variable speed fan, which could produce air velocities bracketing those expected in a full-scale air-cooled condenser. The laboratory tests used a single nozzle centrally located at the entrance to the inlet duct. While changing the length of the connecting pipe nipple at the base of the test stand could vary the distance from the nozzle to the mist eliminator, it was typically fixed at a distance of 6 feet. Testing of the nozzle and nozzle/mist eliminator cooling included basic measurements listed in Table 4-2.

Various test runs used Brentwood Industries CDX80, ME100, and CDX-150 mist eliminators. The final tests employed 6 inches of Brentwood CF1900 counterflow cooling tower film fill (that is, heat transfer media) as a mist eliminator. Researchers tested it to assess the mist elimination characteristics of the fill, and to see if any incremental cooling was obtained from further evaporation of the water droplets after they were captured and sequestered in the fill.

The eliminator was located approximately 6 feet from the candidate spray nozzle and upstream of the variable speed fan. This nominally affords about one second or less residence time for evaporation of the droplets, depending on the volumetric flow rate of the test system. Figure 4-4 shows the location of test measurements schematically.

Table 4-2
Summary of Pilot Test Measurements

Test Parameter	Measurement Location

Spray Water Temperature	At nozzle
Inlet DBT, RH	~2.5 ft above grade
DBT, RH Before Drift Eliminator	Initiated on final tests
Nozzle Pressure and Flow Rate	At inlet to Test Nozzle
Pressure Drop Across Drift Eliminators	Via Magnehelix Gage
Airflow Through System	Via Pitot Traverse at Mist Eliminator

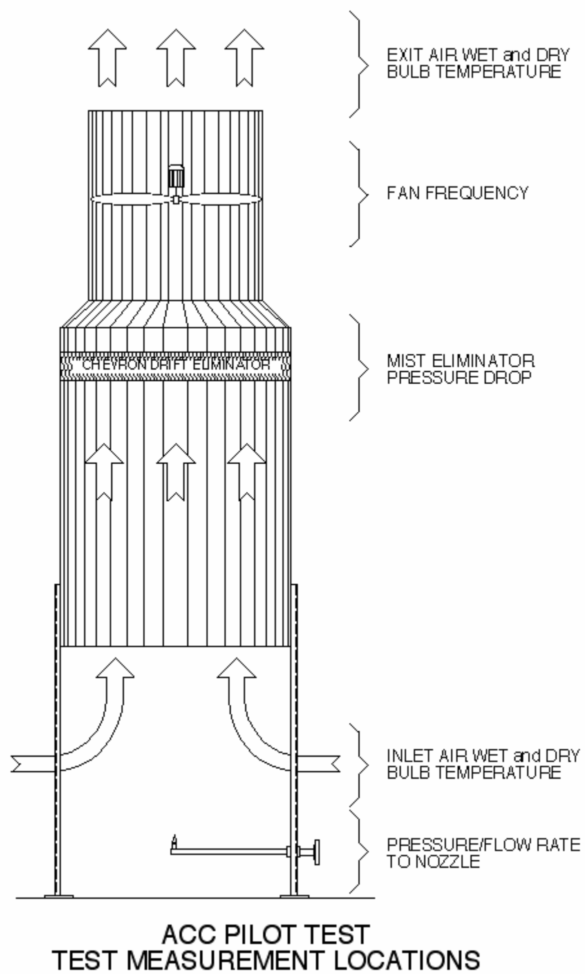


Figure 4-4
ACC Pilot Test, Test Measurement Locations

The presence of unevaporated droplets, which wetted the dry bulb thermometer, made measurement of the exit air dry-bulb temperature difficult. Droplet evaporation correlations suggested that, at the conditions of the tests, there would not be sufficient time to completely evaporate the anticipated distribution of droplet sizes. Initial measurements of inlet and exit psychrometrics were made with a Visalia Dry-Bulb/Relative Humidity instrument. Even with shielding of the instrument, there was some question of the accuracy and consistency of the measurements due to the wetting of the probe. A switch to Cooling Tower Institute-type psychrometers was made in the early part of the pilot tests. Figure 4-5 shows a view of the exit air psychrometric measurements, using the CTI-type psychrometers.

Figure 4-5
View of Exit Air Psychrometric Instruments

Test Results

Baseline tests developed relationships for air and water flows as a function of measured quantities.

Spray Water Flow Rate

Researchers developed spray nozzle flow rates as a function of supply pressure for each of the candidate nozzles. In addition to an in-line flow meter, they performed "bucket/stopwatch" type tests. Figure 4-6 provides an example correlation, indicating the expected second-order relationship between pressure and flow.

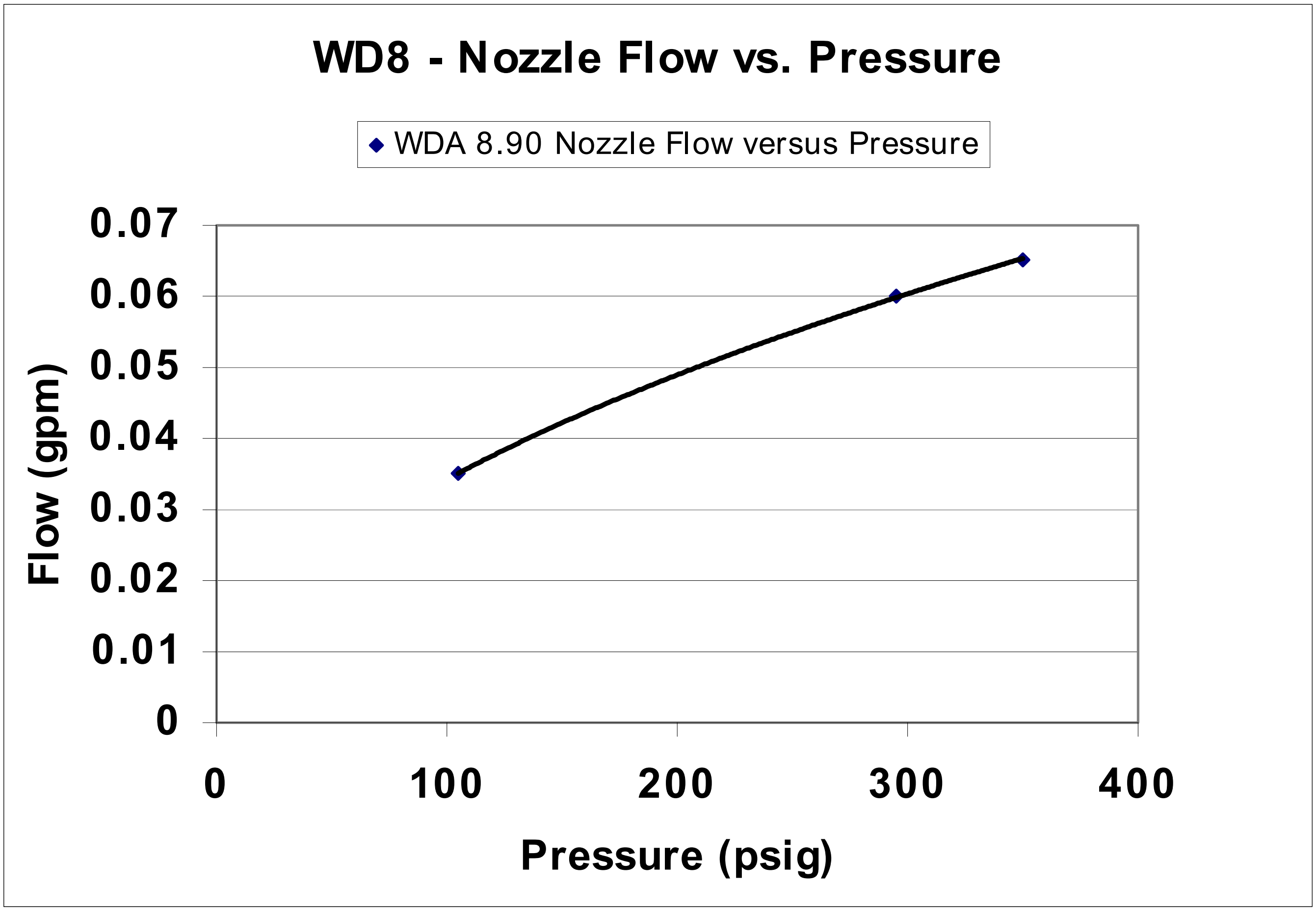


Figure 4-6
WD8 – Nozzle Flow vs. Pressure

Airflow Measurements

Testers performed airflow measurements using a pitot-static tube and inclined manometer. They established equal area points upstream of the mist eliminators in the four-foot diameter section of the pilot test facility. Appendix A contains a representative airflow data sheet. Figure 4-7 is an example of the measurements.

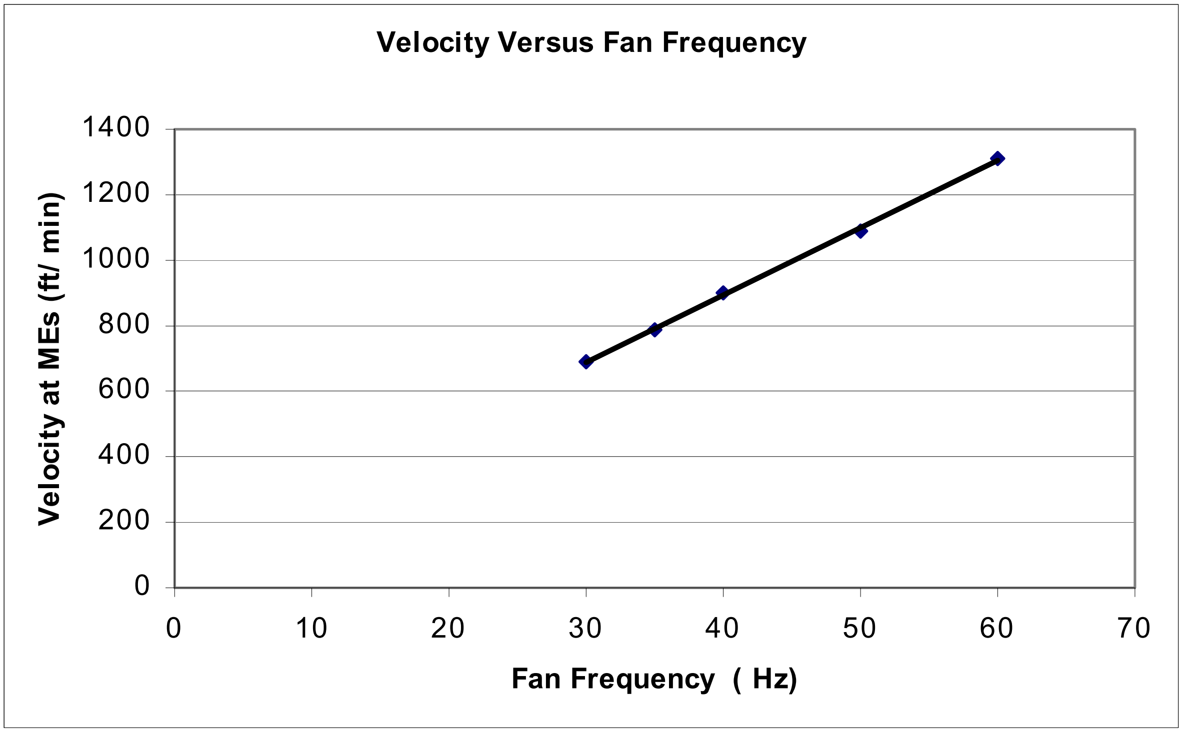


Figure 4-7
Velocity Versus Fan Frequency

Evaporative Cooling Effect

Appendix A contains the complete set of data obtained on the cooling effect of the several nozzles at various water and airflow rates and nozzle locations. The data include air and water flow rates, inlet and exit air dry and wet bulb temperatures, and pressure drop across the mist eliminator. The last column headed $\{(DB_i - DB_0)/(DB_i - WB_i)\}$ is the Effectiveness Ratio.

Test Results

Figures 4-8 and 4-9 present the results of the tests. Figure 4-8 plots the temperature reduction achieved plotted against the product of the wet bulb depression (a measure of the driving force for evaporation) and the spray rate (related to the droplet surface area). While the results exhibit a large amount of scatter, there is a general pattern of increasing cooling effect rising to a plateau or maximum corresponding to a practical limit of 65 to 75% of the wet bulb depression.

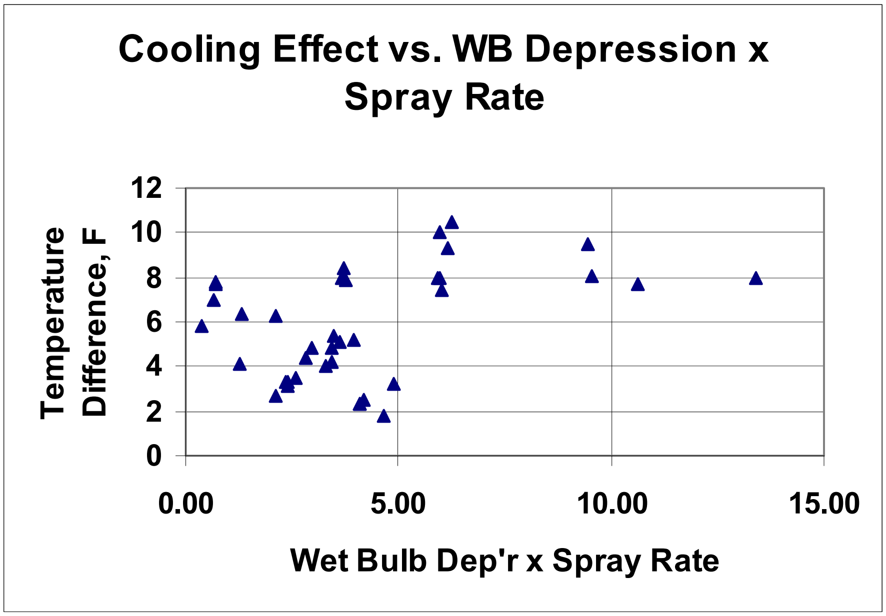


Figure 4-8
Cooling Effect vs. WB Depression × Spray Rate

Figure 4-9 separates the data by nozzle type in a plot of effectiveness (as defined above) vs. wet bulb depression. There is no discernible distinction between nozzle types. However, individual nozzles of both types can achieve over 70% effectiveness under some conditions.

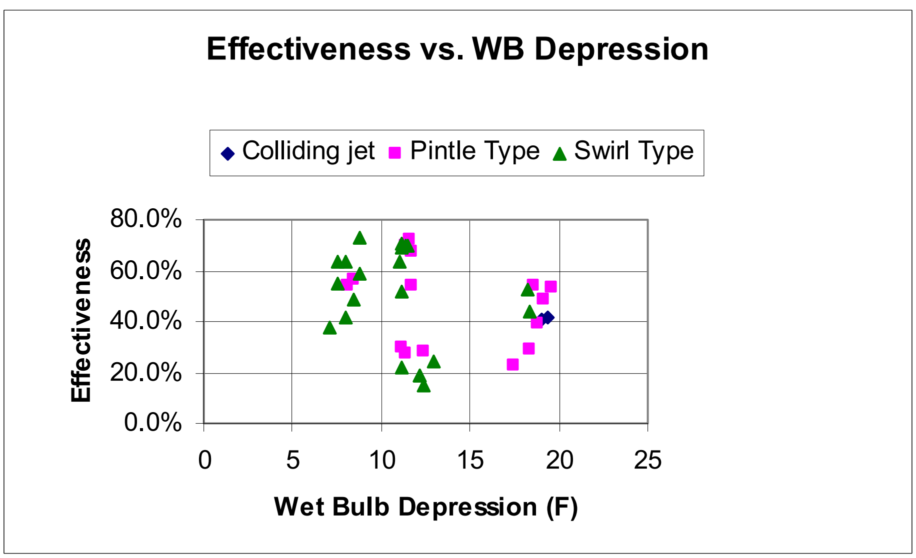


Figure 4-9
Effectiveness vs. WB Depression

Additional observations on the details of the test results follow, leading to a recommendation of three nozzles to be tested at larger scale.

Effectiveness Ratio

The purpose of inlet air-cooling is to reduce the air temperature to a significantly lower level. The theoretical limit of cooling is the adiabatic saturation temperature, which is simulated by the wet-bulb temperature. The effectiveness of any spray cooling approach depends upon a number of factors including, but not limited to the following:

- The size distribution and residence time of the droplets being sprayed
- The prevailing wet-bulb and dry-bulb temperatures

A threshold amount of water is required, which once evaporated, will theoretically cool the air to a level approaching the prevailing wet-bulb temperature. Any amount less than this level or any spray that is lost to gravitational settling or collection on objects, such as structural members or fan blades, will reduce the ability of the spray system to reach the prevailing wet-bulb temperature. Further, if the spray water temperature is greater than the wet-bulb temperature, as was the case in the Pilot test studies, the equilibrium temperature of the exit air will be otherwise higher. The Pilot test data were analyzed for “effectiveness” as defined by:

$$\textbf{Effectiveness} = \frac{\textbf{(Dry Bulb at Inlet – Dry Bulb at Exit)}}{\textbf{(Dry Bulb at Inlet – Wet Bulb at Inlet)}}$$

Based on those analyses, the effectiveness of the nozzles, under the prevailing operating conditions, varied from less than 20 percent to greater than 60 percent. While correlations in the variables were scattered, it was noted that effectiveness did vary with residence time in the pilot test cylinder. Data taken on July 18, 2001 with a low-flow nozzle indicated that a reduction of the fan frequency and therefore airflow rate caused a precipitous drop in exit air temperature. Further, increasing of nozzle pressure improved the effectiveness as noted in the data acquired on August 7, 2001 on nozzles PJ28 and WDB8. Effectiveness appeared to improve by 10–15 percent when nozzle pressures were increased from 110 to 300 psig. Improved droplet size distribution and dispersion are the probable contributors to the increase in effectiveness.

Mass Balance and Psychrometric Check

Researchers performed mass balances and psychrometric checks on example Pilot Test data. For instance, given the inlet and outlet air psychrometric properties, one could compare the water added to the air via the nozzles, with estimates of water addition from the calculated air properties. Table 4-3 provides an example mass balance. In this case, the apparent increase in air moisture content is somewhat less than would be calculated based on the nozzle flow rate. Clearly, the mist eliminators removed some water, while some collected on and drained from the walls of the test cylinder. Mass balance comparisons varied significantly, in part due to the potential inaccuracies of the exhaust air psychrometric determinations and, to a lesser extent, the wind effects on the sprayed water.

Table 4-3
Example Mass Balance

Example Mass Balance for Nozzle Pilot Tests

	Test of Nozzle PJ-28	
Fan Hz/Air velocity	30 Hz	688 ft/min
Nozzle Press (psig)		300
Nozzle Flow (gpm)		0.32
Inlet Wet Bulb		67.25
Inlet Dry Bulb		86.35
Spec Humidity (lbs H20/lb - air)		0.00995
Outlet Wet Bulb		68.4
Outlet Dry Bulb		76.1
Spec Humidity (lbs H20/lb - air)		0.01317
Air flow rate (lbs/min)		634.2
Increase in Moisture (lbs/min)		2.044
Calculated Liquid Flow (gpm)		0.245
Nozzle Test flow rate (gpm)		0.320
Difference (gpm)		0.075

Additional Observations

A number of conditions and perceptions are worth noting as part of the evaluation of the pilot tests of nozzle performance. While they include some opinions and comparisons based on previous experience by the test personnel, they are noted herein due to their potential impact on the interpretation of the data and development of conclusions.

Exit Air Psychrometric Measurements

The measurement of air psychrometric conditions can be difficult and compromised by the presence water droplets. This is particularly true, but not limited to the cases for which there were no mist eliminators in place. The first three test runs (July 16,17, and 18) used a Vasalia dry bulb and humidity instrument. In the presence of droplets, it is probable that droplets in the exhaust air affected the exposed sensing elements. It is therefore likely that these three sets of tests were less accurate, relative to exit air properties, than the subsequent ones, which employed CTI-type psychrometers. Calculations indicate that a 1° F error in wet-bulb temperature can result in a 25 percent error in the calculated evaporation rate, depending upon the prevailing psychrometric conditions. Errors in dry bulb temperature measurement have less impact on the calculated moisture in the air.

Nozzle Spray Patterns

The character of the spray patterns observed from the nozzles was easy to discern, even for those less experienced in the "art." Pintle-type nozzles had an obvious sector of spray blocked by the pintle itself. Additionally, sporadic large droplets were seen in the flow pattern. These conditions are seen in the previous spray pattern figures. Both the swirl-type and pintle-type nozzles exhibited reasonably uniform spray patterns, even at lower test pressures of 100–200 psig. Testers visually determined that the sprays were expectedly not as fine at reduced nozzle pressures. Notwithstanding the reduction in nozzle flow rate with pressure and the incremental loss (that is, fall out) of some larger drops, it likely that the operation of many of the test nozzles at pressures between 150–250 psig would not substantially compromise the objective of inlet air-cooling in full-scale situations.

Test Cylinder Cooling

The impaction and collection of droplets inside the pilot test cylinder resulted in obvious cooling of the cylinder. The affect of this on the results of the pilot tests is not known. In the case of full-scale extrapolations of this condition, it is likely that this effect would be less due to the reduced surface area for droplets to collect, vis-à-vis the system airflow rate.

Wind Effects

Wind conditions during the pilot tests were often greater than 5–10 mph (estimated). Due to the fine sprays from many of the candidate nozzles, it was easy for these droplets to be carried outside the confines of the pilot test cylinder. Even with windscreens in place, testers observed some "blow through" and loss of water.

Spray Water Temperature

Spray water temperatures were uncontrolled and typically about 90–93° F. The higher spray water temperatures had an impact on the potential cooling effect of the pilot test facility exit air. Estimates of potential impact on exit air dry-bulb are 0.5–1.0° F, depending upon the spray water temperature and airflow rate.

Summary and Conclusions

Researchers tested a number of low-pressure (that is, <300 psig) nozzles in combination with three commercially available mist eliminators and a film-type droplet collection media. Test data indicate that significant reductions in the inlet dry-bulb temperatures can be achieved, even when the wet-bulb depression is small (less than 10° F). It is noted that the cooling effects observed may be conservative for the following reasons:

1. Sprayed water temperatures averaged about 91° F, while entering air wet- and dry- bulb temperatures were about 60–70° F. This may have increased exit air dry-bulb temperatures by up to 1.0° F above what might have been obtained with spray water at ambient temperature.

2. Depending upon the spray nozzle tested, some of the water hit the inner surface of the test cylinder and drained to the concrete pad below. As such, this water probably contributed little to the cooling of the inlet air.

The test results yielded the following conclusions:

- Tested mist eliminators indicated exceptional performance in eliminating droplet carryover before the fan. No droplet carryover was felt for drift eliminator face velocities less than about 1300 fpm. The Brentwood CF1900 film pack also exhibited good droplet removal capability at face velocities of over 1000 fpm. However, despite the intuitive appeal of a comparatively low-pressure drop, film-type heat transfer media for droplet interception and additional cooling, the data did not show any exceptional cooling. Unfortunately, as was the case with many of the tests, the wet-bulb depression was low during the period of evaluation.
- Confinement and protection of the spray pattern are important considerations for minimizing water losses from wind effects. If seen on a pilot-scale basis, it is also, no doubt, an important consideration for full-scale inlet air-cooling systems.
- Decreasing the air velocity (and increasing the evaporation time) leading to the mist eliminator increases reductions in dry-bulb temperature as expected. Increasing the nozzle pressure also enhances them.
- Evaluation of water use and mass balance calculations as compared to the measurements of those quantities will require more investigation and analysis.

Recommendations for Field-Testing

Both pintle- and swirl-type nozzles were recommended for use in the full-scale phase of testing at Crockett. The pintle-type nozzles exhibit what appears to be a somewhat finer spray pattern than the swirl-type nozzles, even though there is sporadic generation of large droplets. The swirl-type nozzles are capable of generating a fine spray at higher flow rates per nozzle. This makes them an attractive choice for full-scale applications, from a capital cost and maintenance cost perspective.

In making recommendations for the nozzles used for the Crockett testing, the following additional factors came into play:

- How many nozzles are needed and can be accommodated by the spray rack, in order to have the required 20–25 gpm total spray rate for cooling of the air to a single cell?

- What is the availability and delivery of candidate pintle-type and swirl-type nozzles? Further, what modifications are needed to the nozzles in order to accommodate the nozzles and optimize their performance?

The Crockett tests used two pintle-type nozzles (the PJ28 and the PJ20) and one swirl-type (the WDB14-90) .

Laboratory data leads to the following conclusions:

- Tests included a number of low-pressure nozzles (that is, <300 psig) in combination with three commercially available mist eliminators. Test data indicate that significant reductions of inlet dry bulb temperature can be achieved, even when the wet-bulb depression (that is, the difference between ambient dry-bulb and wet-bulb temperatures) is small (that is, less than 10° F).
- Tests of mist eliminators indicated exceptional performance in eliminating droplet carryover before the pilot test fan. There was no apparent droplet carryover (based on simple visual and hand exposure assessments) for drift eliminator face velocities less than about 1300 fpm.
- Confinement and protection of the spray pattern are important to minimize water losses from wind effects.
- Decreasing the air velocity (and increasing the evaporation time) leading to the mist eliminator enhances reductions in dry-bulb temperature as expected.
- Evaluation of the water use and mass balance calculations as compared to the measurements of those quantities will require more investigation and analysis.

5
FIELD TEST PROGRAM

Introduction

Researchers conducted field-testing of the spray cooling enhancement concept on a full-scale air-cooled heat exchanger at an operating electric power generation plant. This section presents a description of the field test program including the following:

- The host plant and the air-cooled equipment
- The spray delivery set-up
- The instrumentation and data acquisition system
- The measurements taken
- The reliability and consistency of the measurements
- The effects of uncontrolled variables (wind and ambient temperature)
- The effect of the spray itself on the measurement reliability

Test Site

Researchers conducted field tests on a single cell of a full-scale air-cooled heat exchanger at the Crockett Cogeneration Plant located in Crockett, California, USA. The site is located on the south shore of the Carquinez Straits, approximately *25* miles northeast of San Francisco.

The plant, which began operation in 1996, is a 240 MWe gas-fired, combined-cycle unit (GE, STAG 107FA), equipped with a 160 MW gas turbine (GE, Model 7FA), a heat recovery steam generator (HRSG) (Henry Vogt Co.), and an 80 MW steam turbine. The turbines are connected to the generator on a single shaft. The plant supplies electricity to the grid and provides process steam (nominally 250,000 lb/hr @ 450 psi) to the C&H Sugar Company's refining plant located on a neighboring site.

The site meteorology is favorable to dry cooling. The temperature is normally cool, with an annual average of about 65° F. There are, however, three to four hundred hours per year at 80° F or above, providing many opportunities to benefit from the use of enhancement methods.

Cooling System

Two air-cooled exchangers, colocated as a single unit on the roof of the turbine building and just east of the HRSG, provide plant cooling. Figure 5-1 shows the layout of the air-cooled units.

Air-Cooled Condenser (ACC)

A Balcke-Dűrr (BDT Engineering) air-cooled condenser, consisting of 12 cells (A-frame modules) arranged in three rows of four cells each, condenses the turbine steam. Three of the four cells in each row (nine total) are condensing modules; the fourth (three total) is a “reflux” (or dephlegmator) module for additional condensation plus the removal of non-condensable gases.

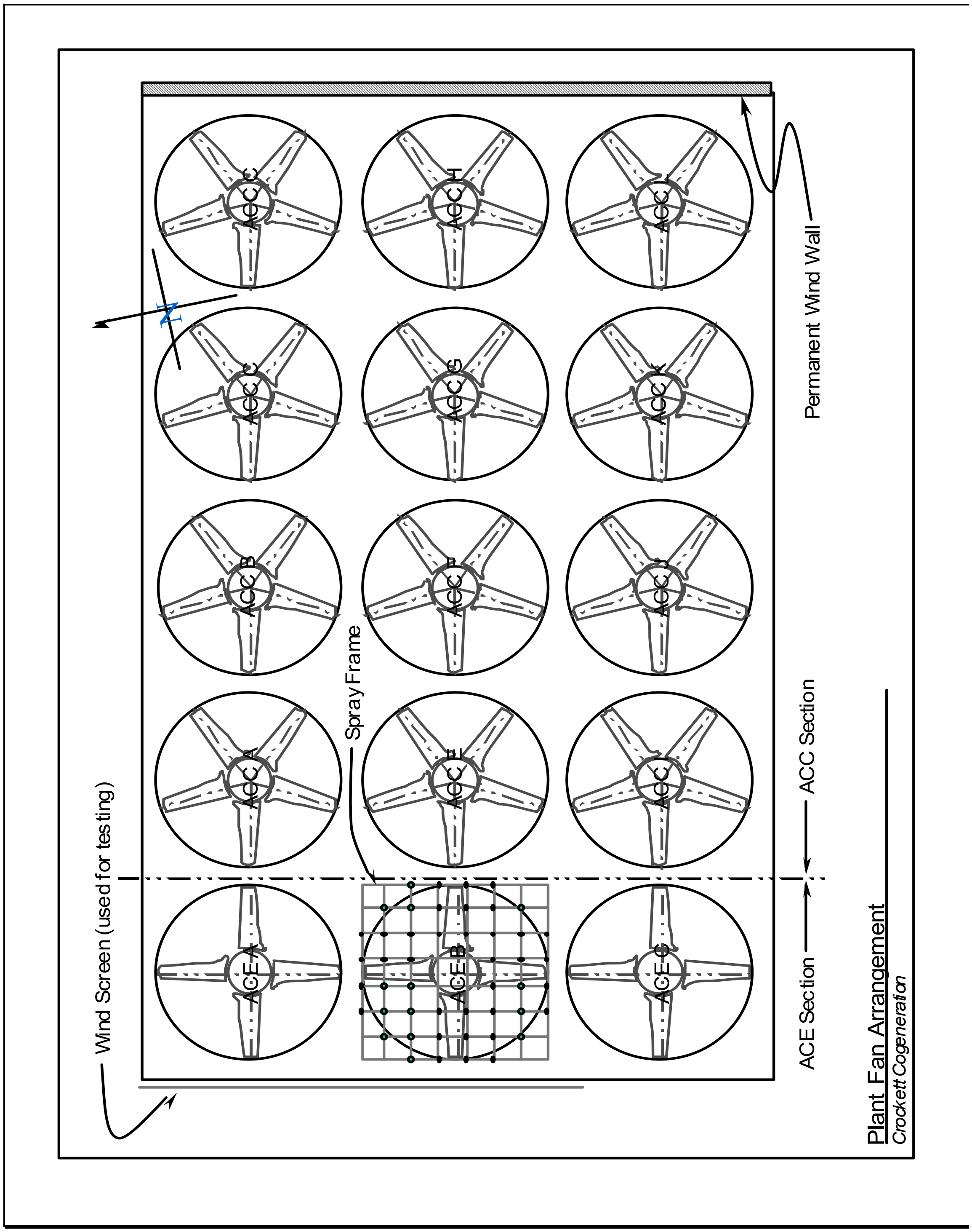


Figure 5-1
Air-Cooled Condenser (ACC) and Heat Exchanger (ACE) at Crockett

Air-Cooled Exchanger (ACE)

An air-cooled heat exchanger provided cooling for a variety of components, such as generator and lube oil coolers, a return condenser trim cooler, and others. The ACE consisted of three A-frame modules, similar in appearance to the ACC cells and located at the west end of each of the three ACC rows. Each cell is capable of handling one-half of the design auxiliary cooling load of 26 million Btu/hr (three 50% capacity units). The center cell (designated by the plant as “ACE Cell B”) was the test cell.

The coolant (water/10% propylene glycol) flow was 5,400 gpm. The unit was designed to maintain the coolant temperature below 126° F at summer design conditions (96° F dry-bulb; 68° F wet-bulb).

Axial-flow, 28-foot diameter Alpina low-noise fans with a specified air flow of 964,000 ACFM per cell supplied cooling air. Each cell consisted of eight finned-tube bundles (four on each face). The tubes were arranged in three staggered rows and connected in a two-pass, tube-side circuit with up-flow in the two outer rows and down-flow in the one inner row.

Test Set-up and Instrumentation

Researchers conducted single-cell tests by spraying water into the inlet air of the middle cell of the ACE, upstream of the fan. They then measured the performance of the cell with and without spray enhancement along with relevant water utilization parameters.

Prevailing winds were from the west and of moderate (5 to 10 mph) strength. However, interference from neighboring structures and induced flow by the air-cooled condenser fans can produce strong and sometimes erratic “localized” wind patterns on the floor of the air-cooled equipment underneath the fans. For this reason, researchers mounted the spray nozzles on a rack, which could be raised to just beneath the test cell fan inlet and hung a windscreen on the windward end of the air-cooled exchanger. Figure 5-1 shows the location of the test cell and windscreen

Spray Circuit

Researchers constructed a frame from Schedule 80 PVC pipe and fittings to support an array of spray nozzles. The frame was approximately 26’ x 26’ to fit within the perimeter of the fan inlet shroud. The frame could accommodate fifty to one hundred nozzles mounted on it in a rectangular array. They were connected with pressure hose in eight rows (with seven equally spaced fittings to which one or more nozzles could be attached). Researchers independently valved each row for step-wise control of the spray flow rate and nozzle distribution across the fan inlet.

Researchers maintained supply pressure at or below 300 psi. They could vary the flow rate, depending on the supply pressure and the number and type of nozzles, from 2 to 20 gpm.

The spray water was high-purity, deionized condensate. The temperature of the condensate varied from 75 to 125° F depending on plant operation. The spray circuit possessed an inline 100µ stainless steel strainer filter, and each nozzle was fitted with a 100µ screen.

Researchers suspended the nozzle rack from the fan shroud support beams with cables and pulleys. They could change the elevation from floor level (approximately 30 feet below the fans for maintenance and nozzle installation) to just below the fan inlet screen with the nozzle exits approximately 2 feet below the fan blades. Figure 5-2 shows the spray rack with the nozzles in operation. Testing included three different nozzle types.

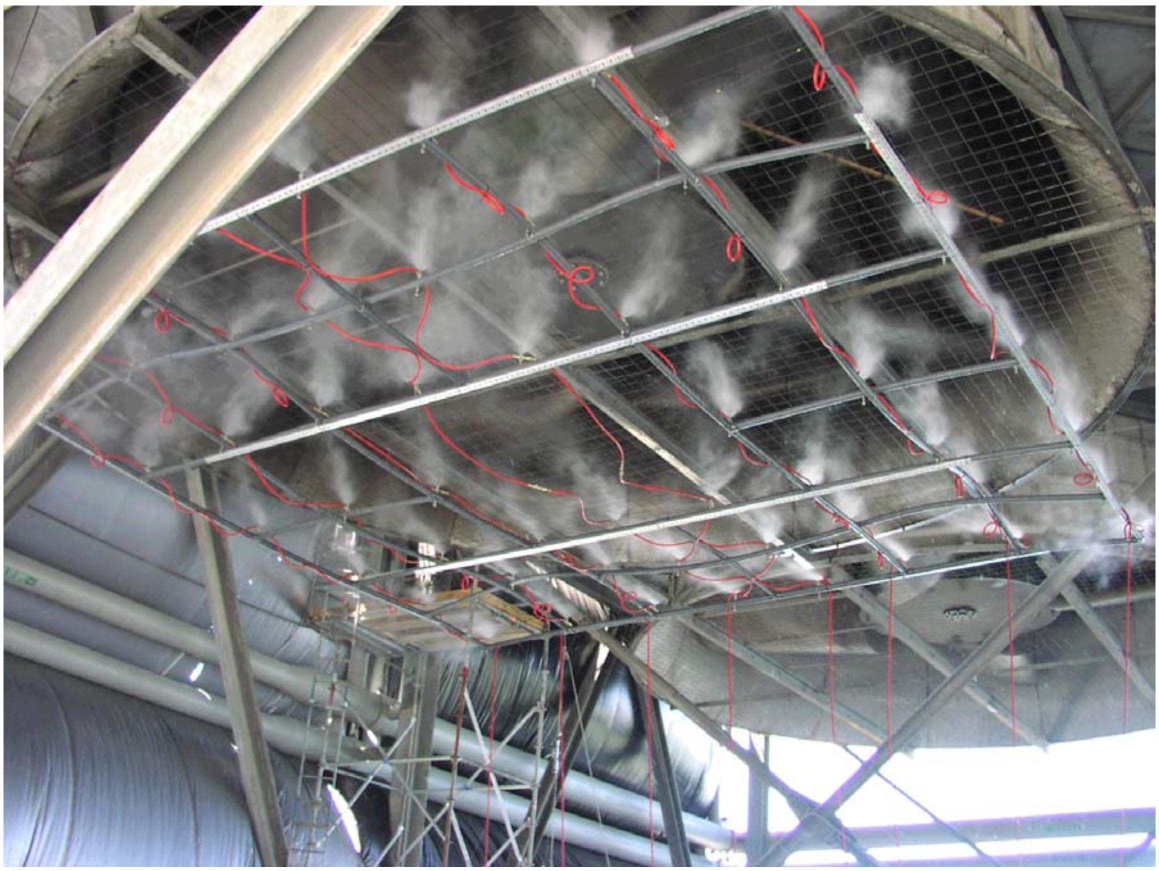


Figure 5-2
Spray Rack and Nozzles in Single-Cell Test

Table 5-1
Nozzles for Single-Cell Tests

Nozzle	Type	Flow/nozzle (gpm) (@ 300 psia)	Sauter Dia. μ
A	Impingement pin (PJ 28)	0.36	45
B	Impingement pin (PJ 20)	0.18	35
C	Internal swirl (WDB 14-90)	0.25	38

Single-Cell Instrumentation

The following measurements were made.

Spray Circuit

- Flow rate, total (Orange Research in-line gauge; 0 to 50 gpm)
- Manifold pressure (Ashcroft 4” gauge; 0 to 500 psi)
- Supply temperature (RTD probe at pump discharge)

Closed Cycle Cooling Water/Glycol

- Coolant flow rate, velocity measured in each of two 10” inlet downcomers; Ultrasonic velocimeter
- Coolant temperature
  - Inlet: from plant control room (measured at discharge of ACE circulating water pump)
  - Exit (ACE): from plant control room (measured in ACE discharge header; mixture of Cells A, B and C)
  - Exit (Cell B): RTD probe (measured in both north and south face tube bundle exit headers)

Air Side

- Air flow, measured at exit of finned tube bundles with 9” propeller anemometer (64 equal-area points on each face)
- Temperature (dry and wet-bulb)

- Inlet: four dry and wet-bulb aspirating psychrometers at floor level (30 ft. below fans); six dry-bulb psychrometers on scaffolds at SW and NW corners of Cell B (10, 20 and 30 feet above floor)
- Above sprays: 12 dry-bulb RTD probes 3 to 5 feet above fan (3 in each of four quadrants at equal-area points); 4 wet-bulb psychrometers 8 to 10 feet above fan (one in each quadrant)
- Exit: eight dry-bulb RTD probes at equally spaced vertical locations one foot from tube bundle exit plane on south face; can be traversed across width of cell

Ambient

- Wind speed and direction at NE corner of ACC on roof of Administration Building

Researchers recorded data in a Hewlett-Packard data logger. They scanned each point four times per minute. Researchers computed and displayed individual points and selected averages at one-minute intervals in both tabular and graphical form.

Measurements: Reliability and Consistency

In spite of the numerous and redundant measurement points on the test cell, there were issues of data reliability and consistency which posed difficulty for the analysis of the results. The heat and mass balances on the cell which determine the cooling effect, the amount of spray water evaporated, and the absolute accuracy of the results depend on the accurate determination of small differences between inlet and exit dry and wet-bulb temperatures. The intrusion of a hot air stream from the neighboring turbine hall, the operation of fans on neighboring cells, variations in wind speed and direction, and transient effects including variable ambient conditions and the response time of the ACE itself affected uncertainties in these values .

Effect of Fan Operation on Inlet Air Temperature

The proper measurement of representative inlet conditions is critical to determining the performance of the heat exchanger and the achievable degree of enhancement from the inlet sprays. Wind and airflow patterns upwind of the exchangers complicated this measurement. Heat sources from the HRSG and a turbine cooler exhaust duct located immediately to the west of the ACE (directly upwind at most times) produced significant temperature differences and stratification at both the floor level and at the various elevations on the scaffolding beneath the B Cell as Figure 5-3 indicates schematically.

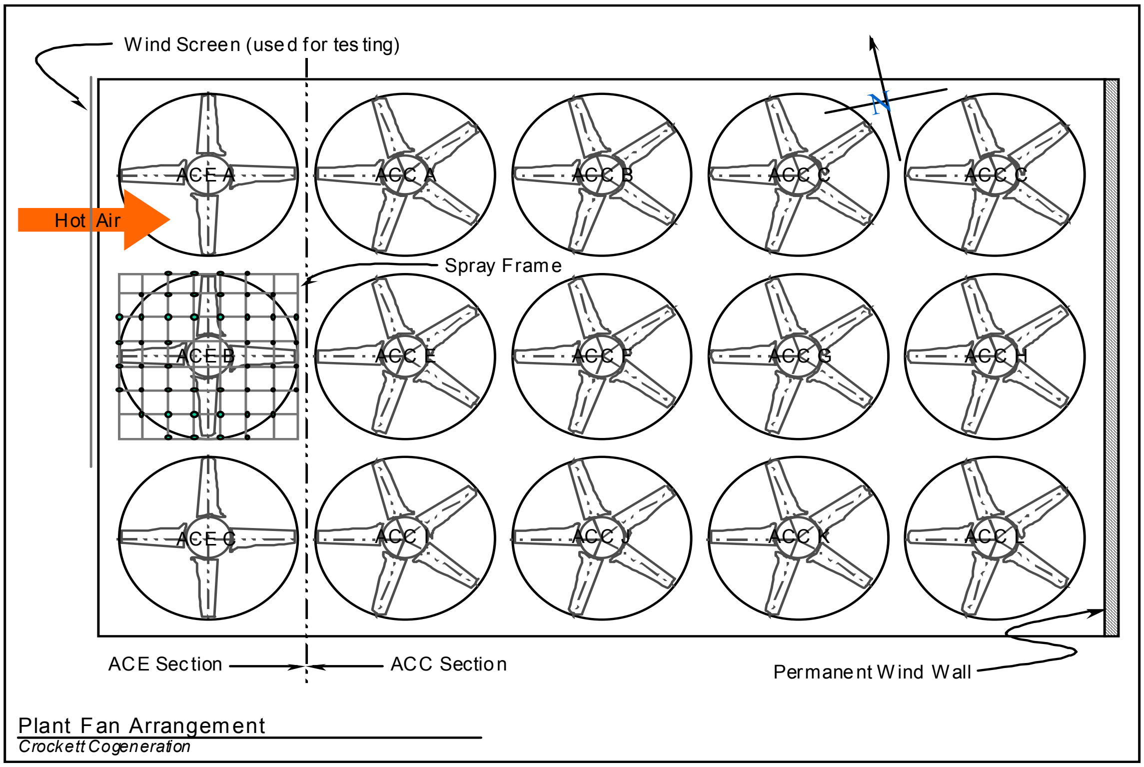


Figure 5-3
Hot Air Stream From Upstream Sources

Figures 5-4 and 5-5 show the spread in the dry-bulb temperatures of the air below the fan (four points at deck level and six on the scaffolds) for different combinations of ACE fans operating. The inlet air steam entering at the northwest corner of the unit (under the A-fan; see Figure 5-1) caused elevated readings of measurements at the north location on the deck level and at all three levels on the north scaffold.

Operating the A-fan, along with the B-fan, intercepts the hot air flow and diverts it into the A-cell. Operation with all (A-, B- and C-fans) fans gave adequate uniformity (~ +/- 2.5° F in the deck level and scaffold measurements). All fan operation also gave excellent uniformity (~ +/- 1° F) in the 16 measurements of the fan exit temperature as Figure 5-6 shows.

In addition, the average values of the fan inlet temperatures at the deck and scaffold levels and the fan outlet temperature are in excellent agreement to within 1° F as Figure 5-7 shows. The behavior of the deck level wet-bulb temperatures (Figure 5-8) and the fan exit wet-bulb temperature (Figure 5-9) is similar, with acceptable uniformity and excellent agreement (within less than 1° F) between the average values during the period with all fans operating (Figure 5-10).

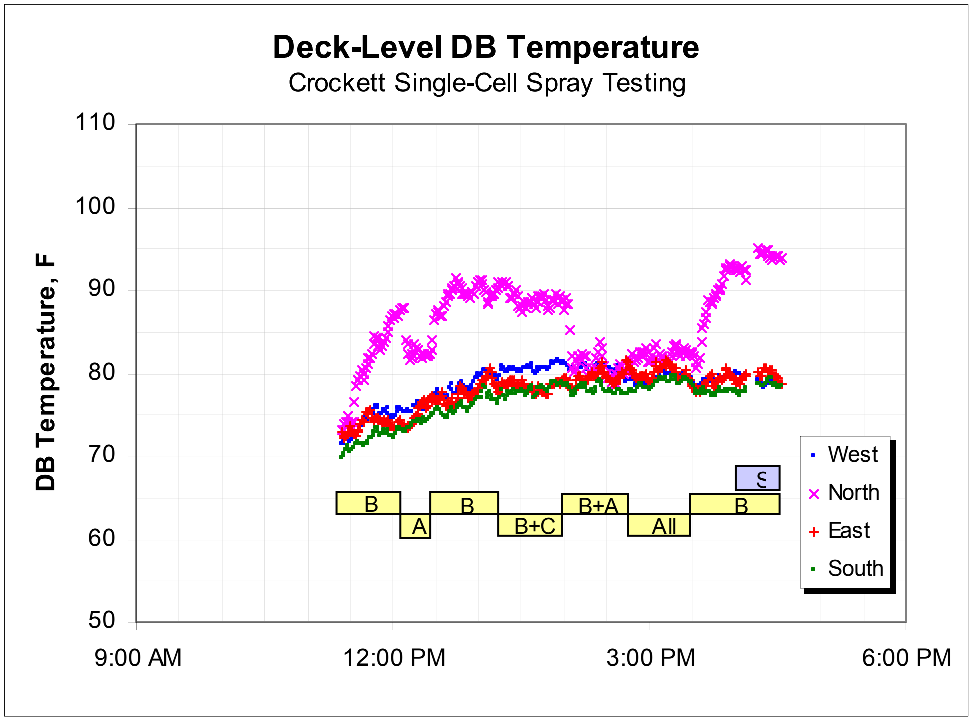


Figure 5-4
Deck-Level DB Temperature

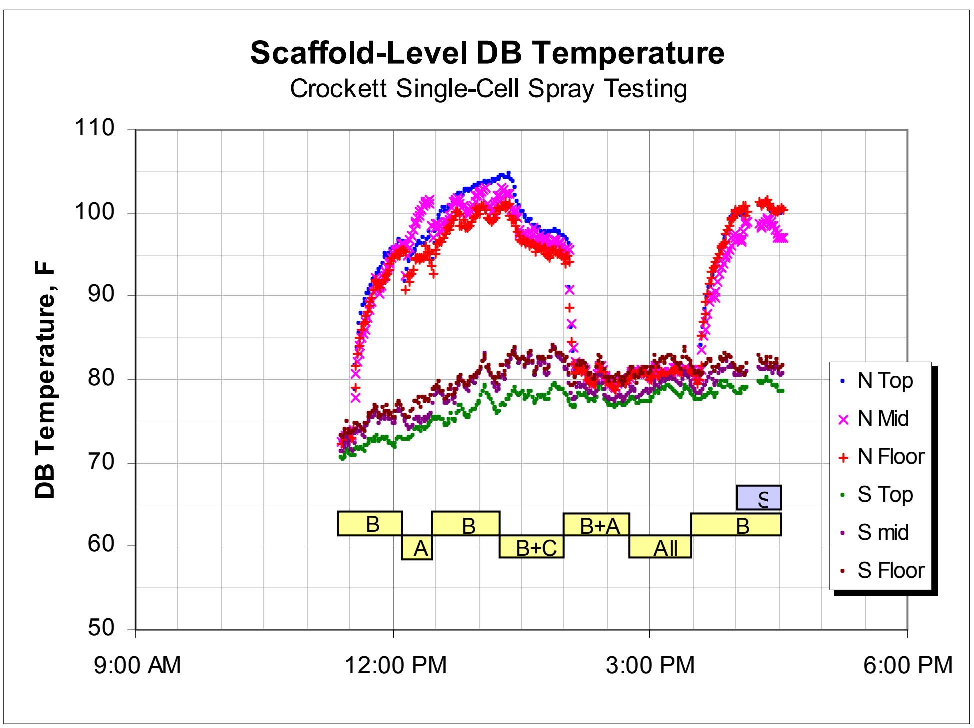


Figure 5-5
Scaffold-Level DB Temperature

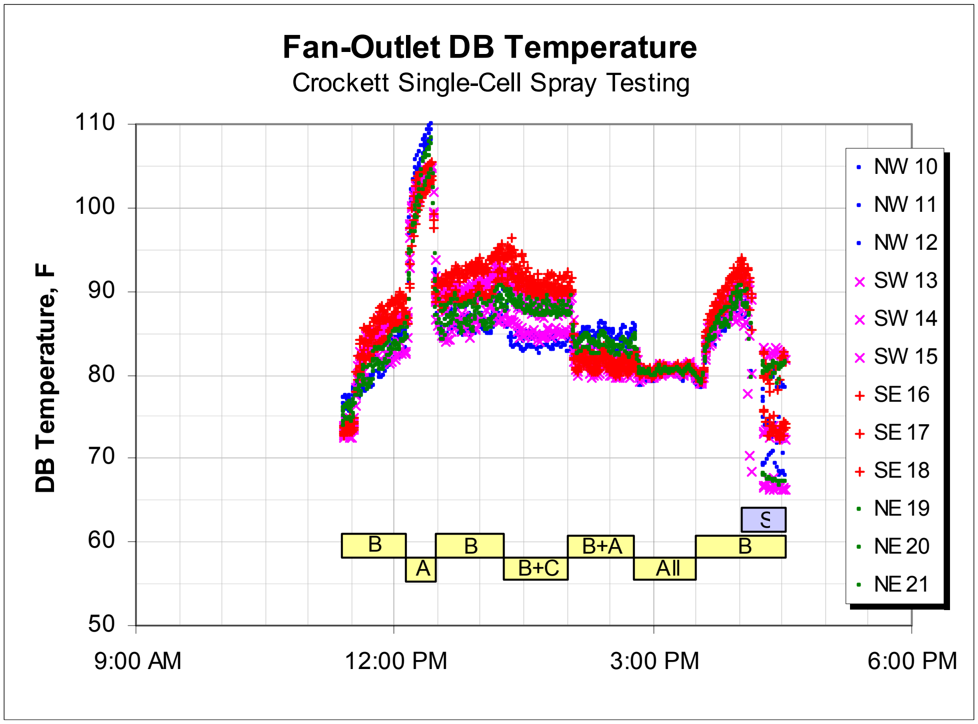


Figure 5-6
Effect of Fans on Inlet Temperature Uniformity

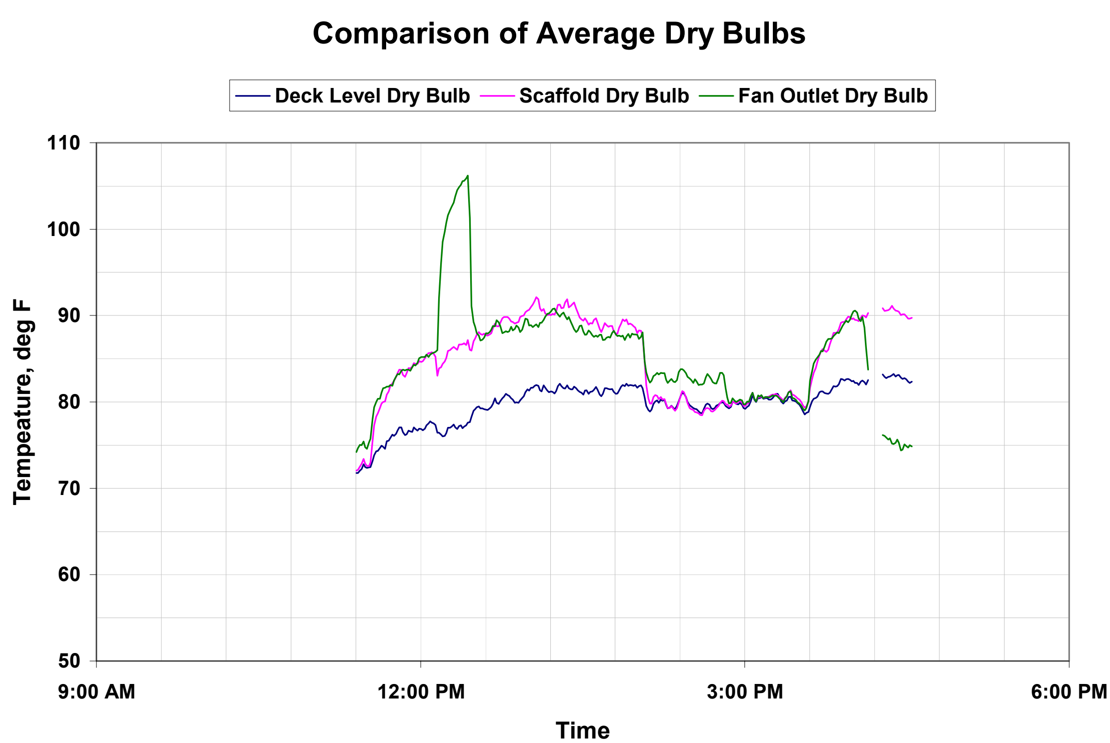


Figure 5-7
Comparison of Average Dry Bulbs

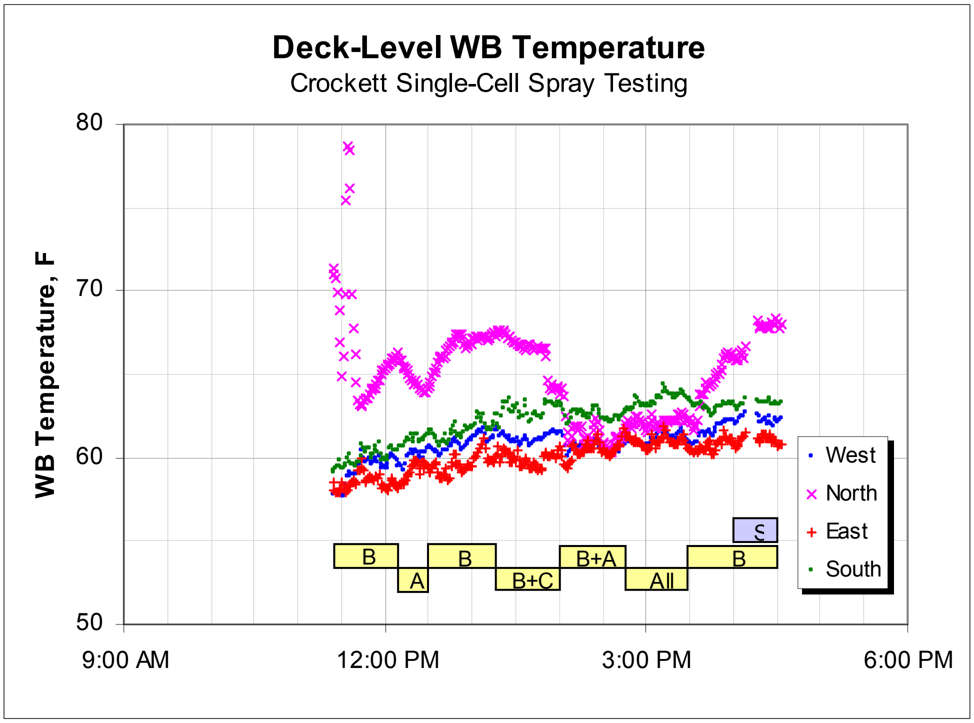


Figure 5-8
Deck-Level WB Temperature

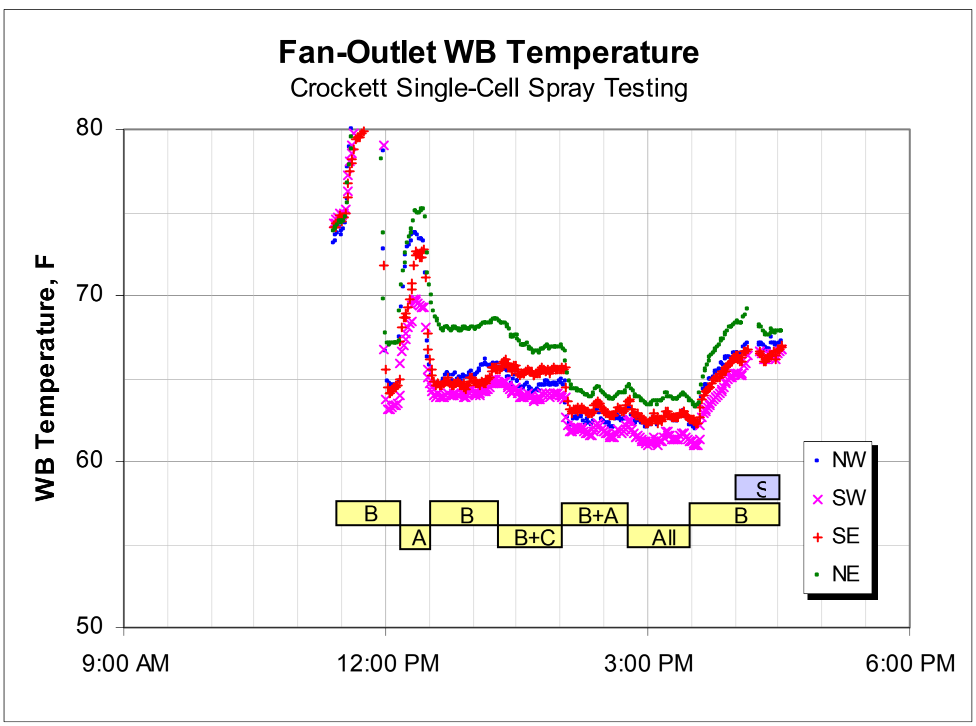


Figure 5-9
Fan-Outlet WB Temperature

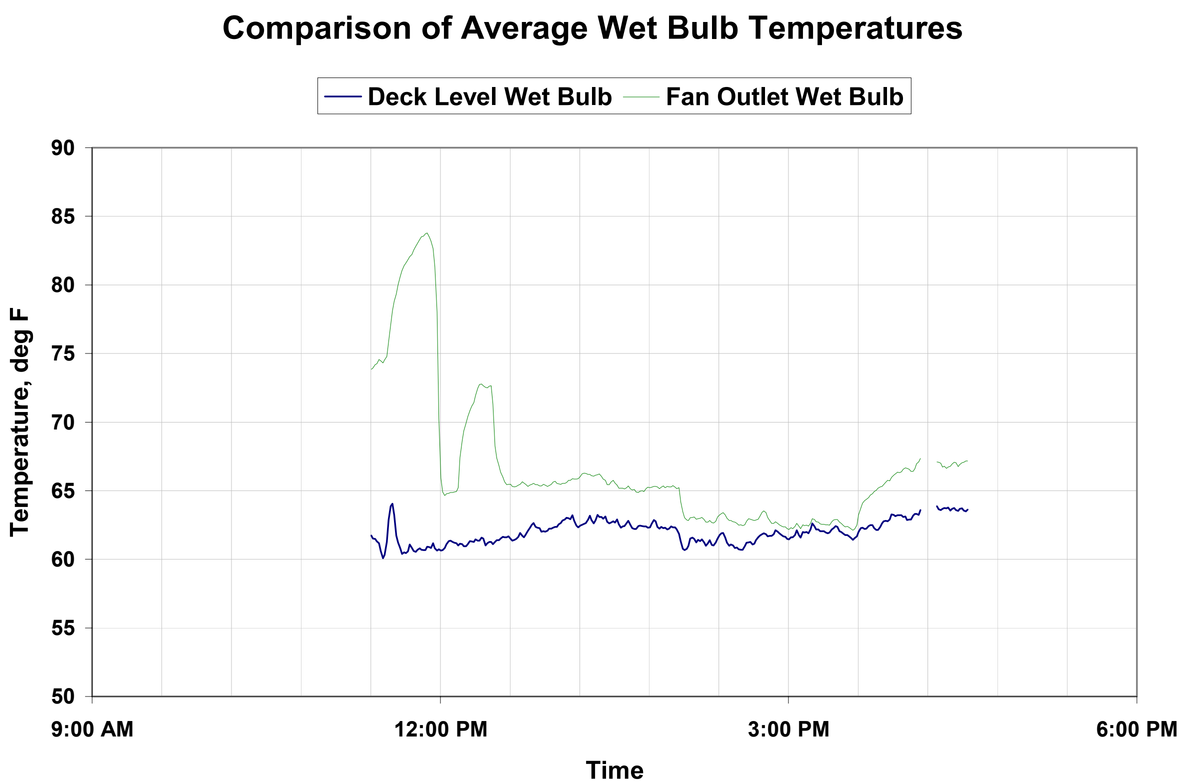


Figure 5-10
Comparison of Average Wet Bulb Temperatures

Temperature measurements at the exit of the ACE finned tube bundles showed larger variation, even under the “all fans operating” condition as shown in Figure 5-11.

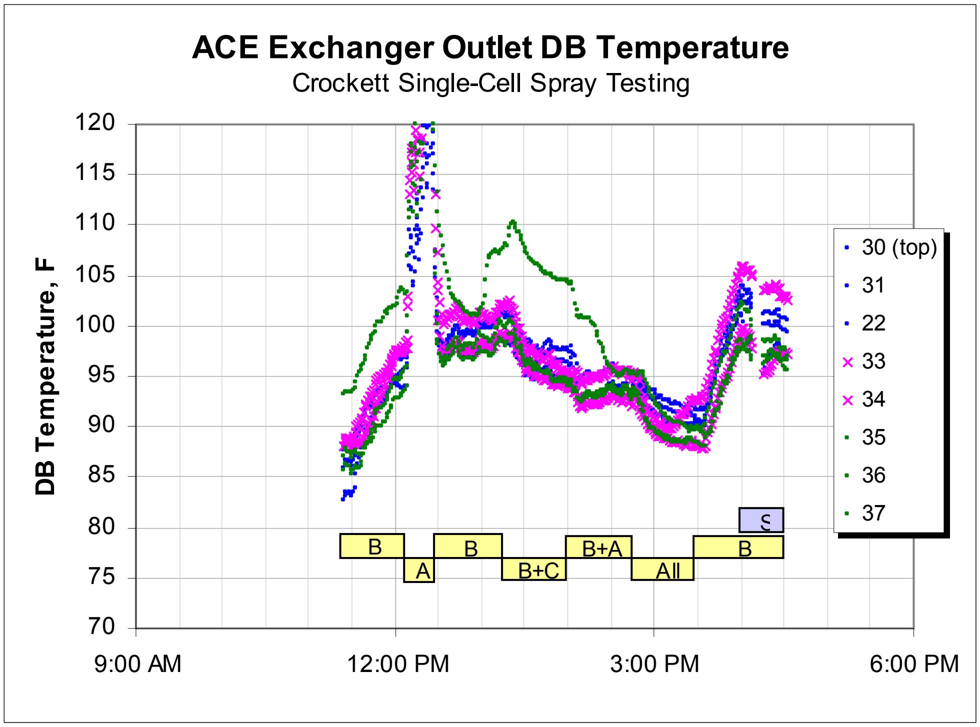


Figure 5-11
ACE Exchanger Outlet DB Temperature

Differences of 6° F from top to bottom (~ +/- 3° F from the average) are typical at that location due to the non-uniformity of the airflow distribution through the tube bundles. Section 6 contains a description of the of the airflow measurement results . The variability among the four measurements near the center of the ACE exit plane is less and the average values gave satisfactory agreement with the water/glycol exit temperature and an acceptable heat balance (see Section 6).

There were two measurements of the water/glycol exit temperature in the exit headers of the north and south faces of the B-cell. They agreed to within 1° F during the "all fans operating" period as Figure 5-12 shows.

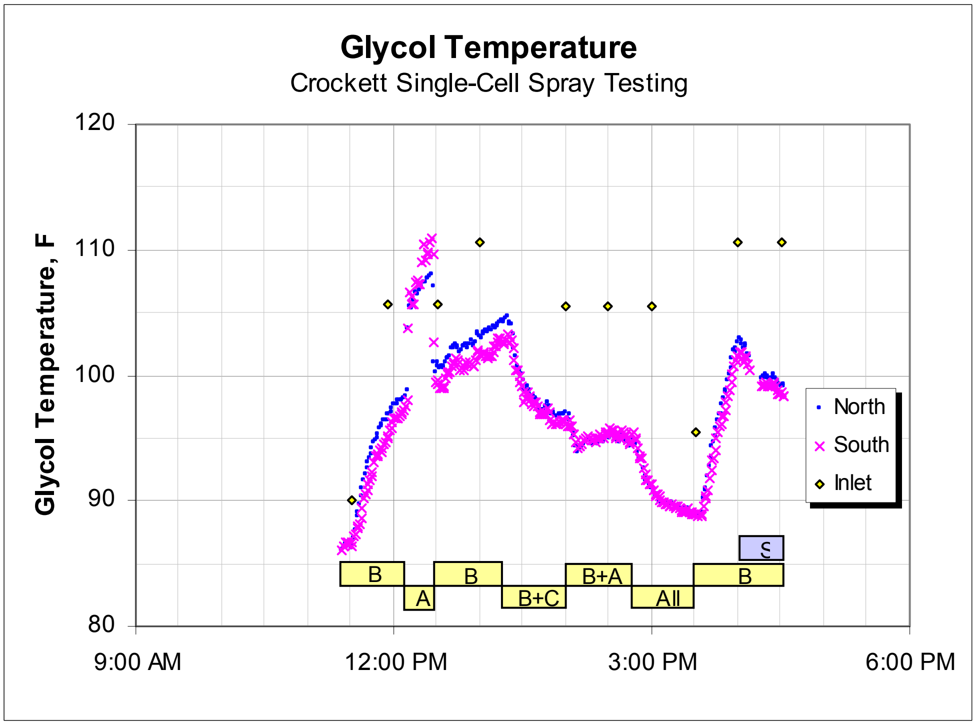


Figure 5-12
Glycol Temperature

Figure 5-13 displays the wind behavior during the testing period. While the readings show considerable short-term variability, the average speed and direction and the general character of the variability remain constant throughout the day. The following section discusses the effect of variable wind behavior on the consistency of the measurements.

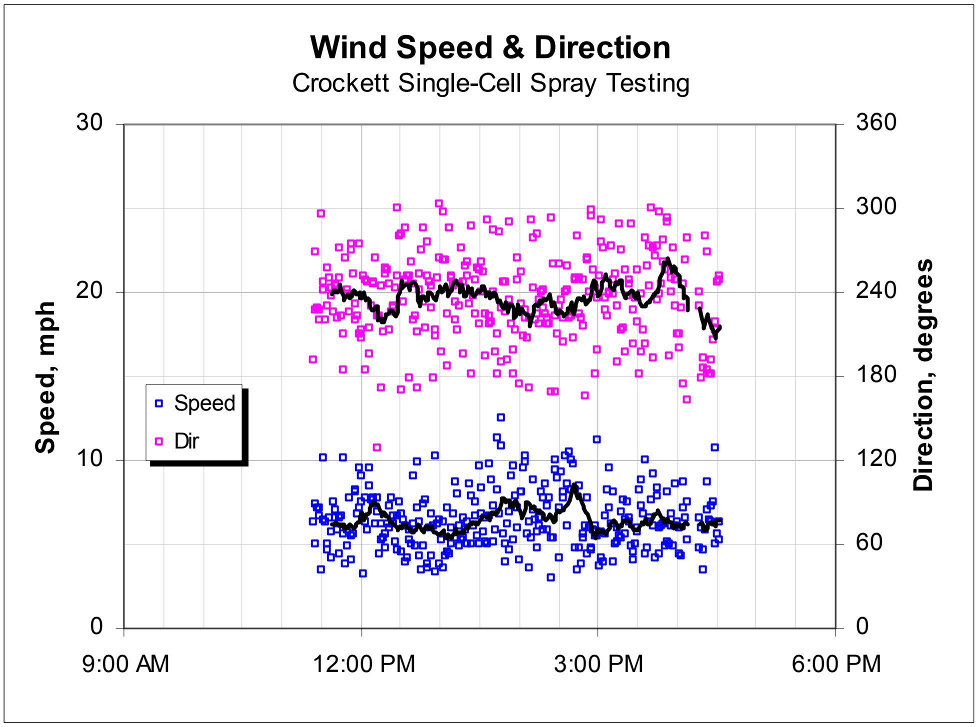


Figure 5-13
Wind Speed and Direction

Effect of Wind

The prevailing winds at the site during the two-month test period were generally from the west (240° to 300° on the plots) at speeds of 5 to 10 miles per hour. These winds from the San Francisco Bay and Pacific Ocean maintain temperate cool conditions at the site for most of the time. Typical ambient temperatures during the test period ranged from 75 to 85° F with wet-bulb temperatures from 60 to 67° F, corresponding to a relative humidity of 30–60%. More rarely, the winds would shift and come from the central valley to the east resulting in hotter, drier conditions. Often there would be relatively sudden and usually short-lived changes in wind speed and direction, which could significantly alter the inlet air conditions temporarily. Figure 5-14 displays the wind conditions during the afternoon of September 7 when, for approximately 30 minutes beginning at about 2:00 P.M., the wind speed increased and the direction, while remaining constant on average, became uncharacteristically steady. Temperature readings during this period departed from their otherwise nearly constant levels; the average inlet dry-bulb temperature at the deck level decreased while the average inlet wet-bulb temperature increased, as Figures 5-15 and 5-16 show. This significantly decreased the wet-bulb depression and hence the driving force for evaporation. Figure 5-17 shows the decrease in the quantity of water evaporated under conditions of constant spray rate.

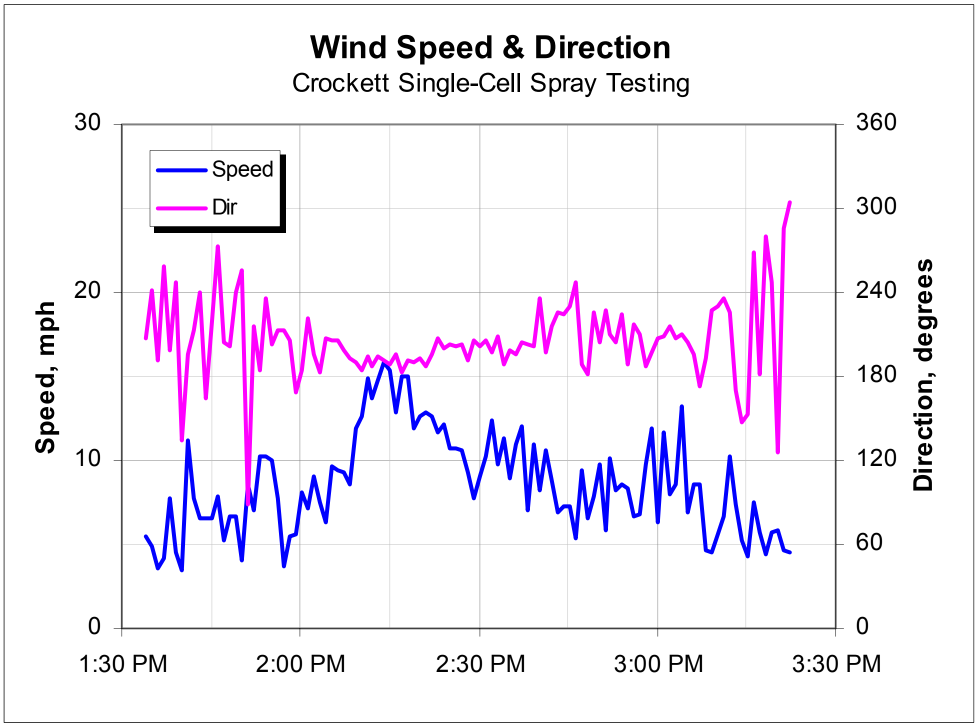


Figure 5-14
Wind Speed & Direction

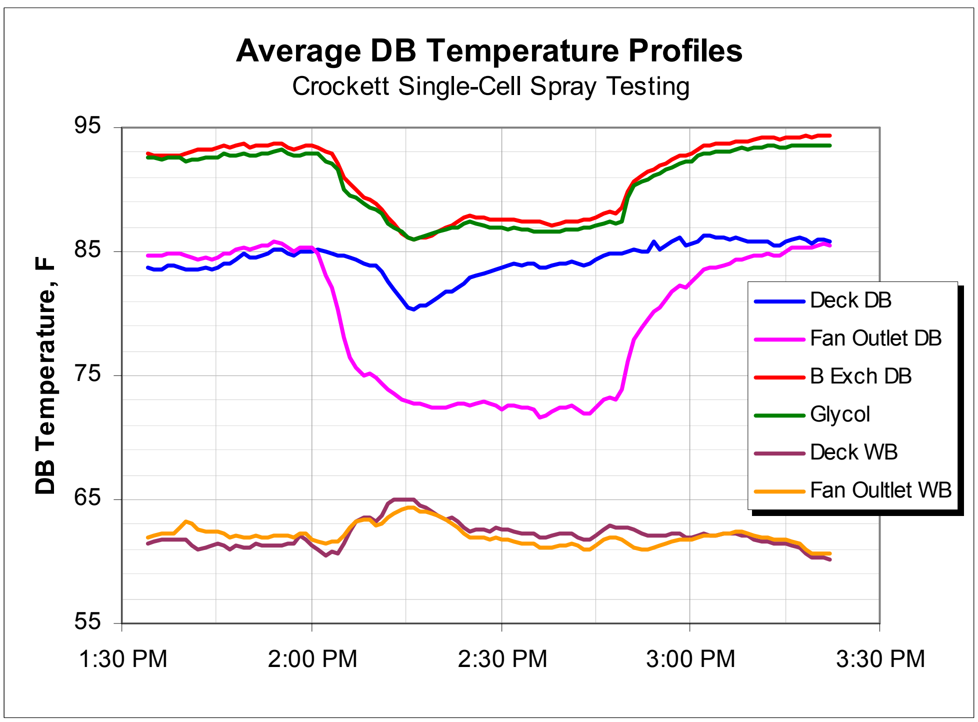


Figure 5-15
Average DB Temperature Profiles

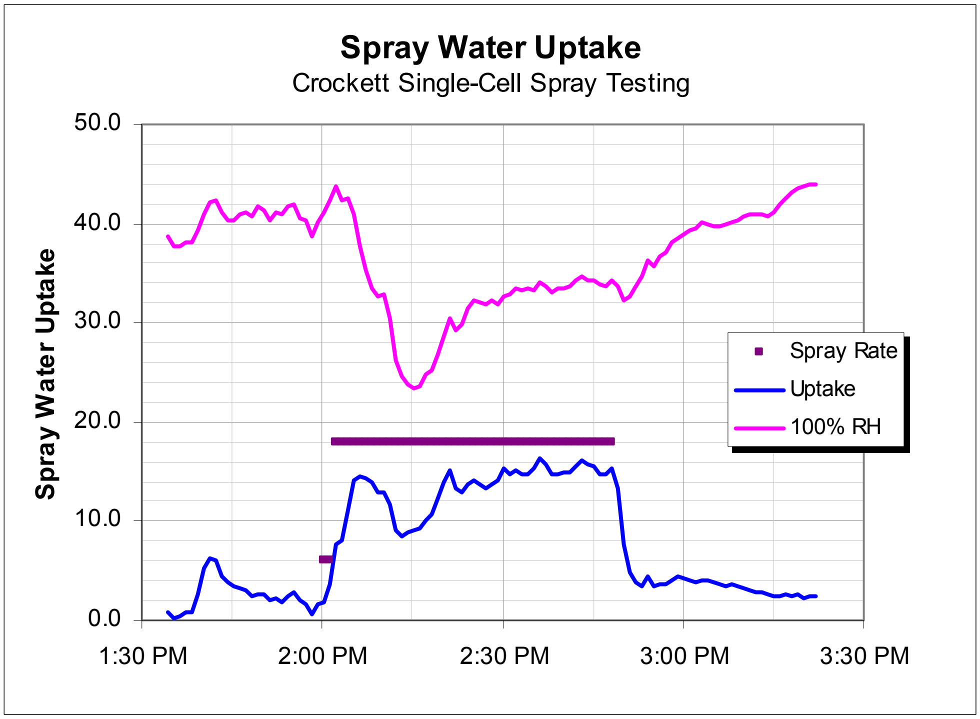


Figure 5-16
Spray Water Uptake

Similar effects can be noted during the tests on October 9 when the wind speed remained steady while the wind direction changed from a steady westerly flow (~ 300°) at mid-day and early afternoon and then exhibited large direction shifts from 1:45 P.M. through 4:30 P.M., as Figure 5-17 shows. Figure 5-18 shows a dramatic non-uniformity in the deck level dry-bulb temperature readings beginning also at 1:45pm; Figure 5-19 shows an oscillatory pattern in the deck level wet-bulb temperature readings, which corresponds roughly to the wind behavior. While the average values of these inlet dry- and wet-bulb temperatures were not strongly affected, ambient changes of this type can make the attainment of steady state conditions very difficult.

Transient Effects

Researchers had to take a number of interacting transient effects into account when interpreting the test results. These include the following:

- Ambient temperature variations
- Heat exchanger response time
- Responses to changes in spray rate

Ambient Temperature Variation

Even under steady wind conditions, the ambient dry and wet-bulb temperatures vary during the duration of individual test runs (typically from late morning to late afternoon). This results in a varying wet-bulb depression, and hence a varying evaporation rate and cooling effect. Figure 5-20 displays the data from September 26, which illustrate this.

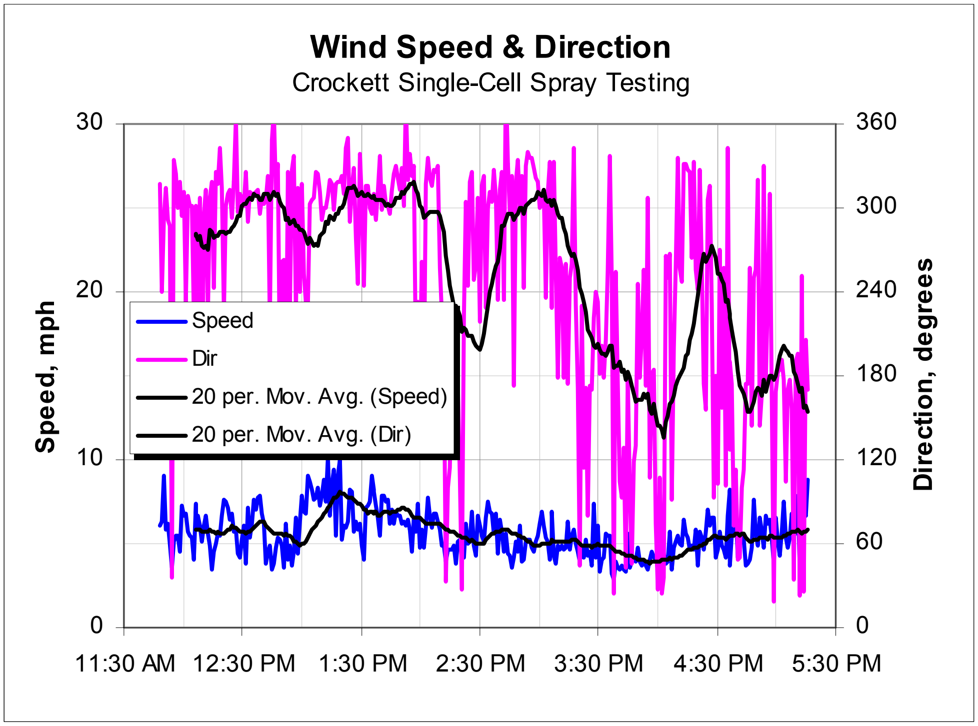


Figure 5-17
Wind Speed & Direction

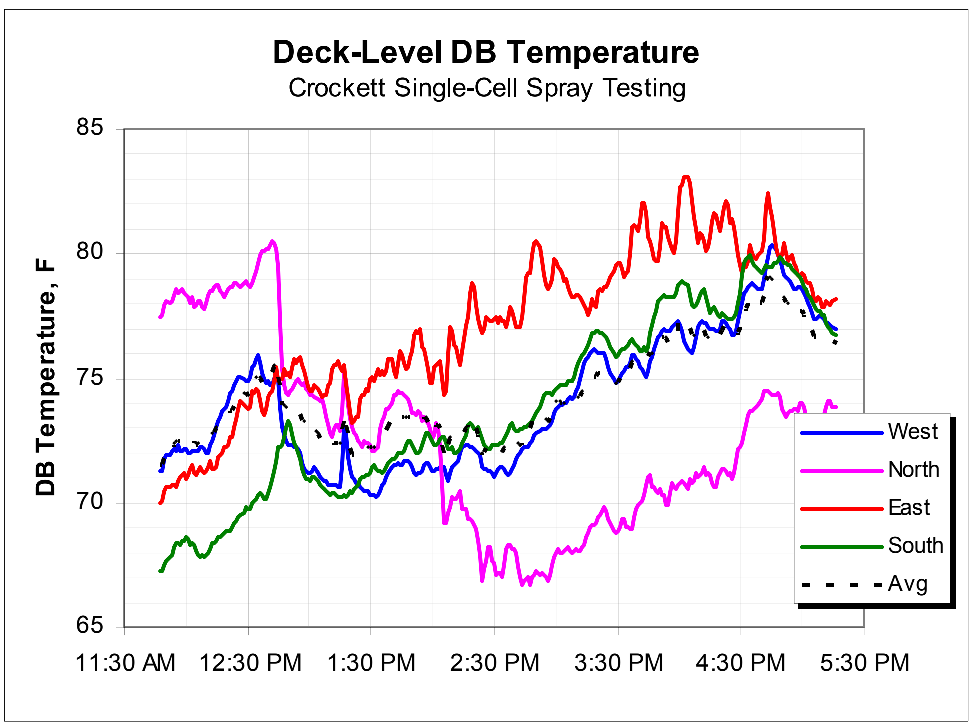


Figure 5-18
Deck-Level DB Temperature

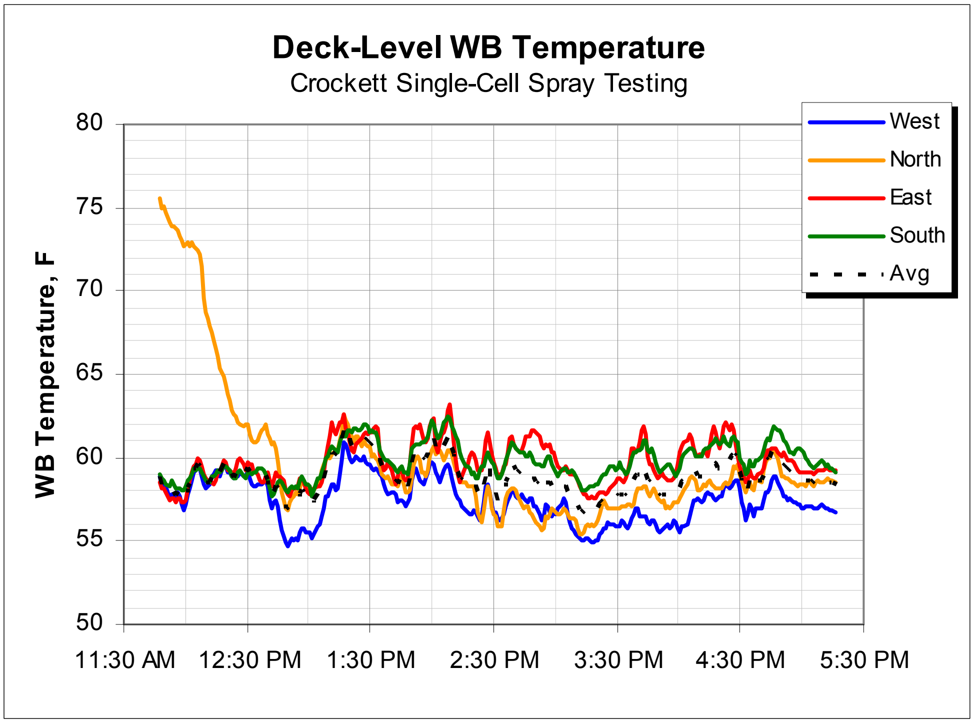


Figure 5-19
Deck-Level WB Temperature

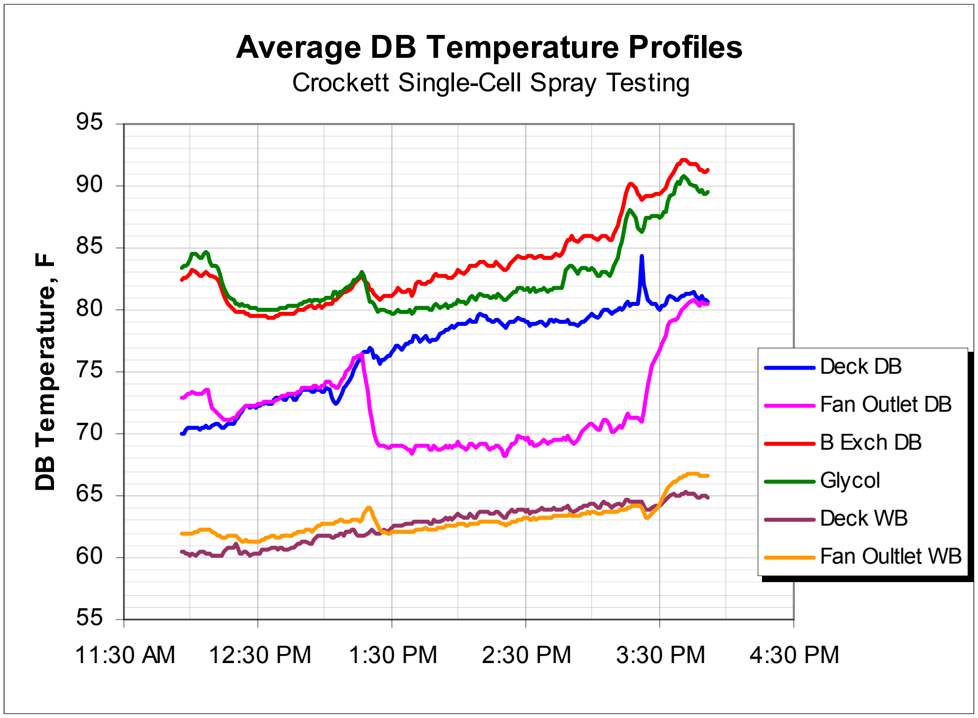


Figure 5-20
Average DB Temperature Profiles

The following observations are noteworthy:

- The average dry-bulb temperature rises from 70° F at noon to 80° F at about 2:15 P.M. and then remains nearly constant until 3:45 P.M. (The spike to 85° F at 3:20 P.M. is a result of a ruptured hose spraying hot water on the north sensor and can be ignored.)
- The average inlet wet-bulb temperature rose uniformly over the entire test period from 60° F at noon to 64° F at 3:45 P.M.
- As a result, the wet-bulb depression increases from 10° F at noon to 17° F at 2:15 P.M. and then decreases to 16° F at 3:45 P.M.
- After the sprays were turned on at 1:15 P.M., the cooling effect rose from 6° F at 1:15 P.M. to 10° F at 2:15 and then leveled off, as Figure 5-21 displays .

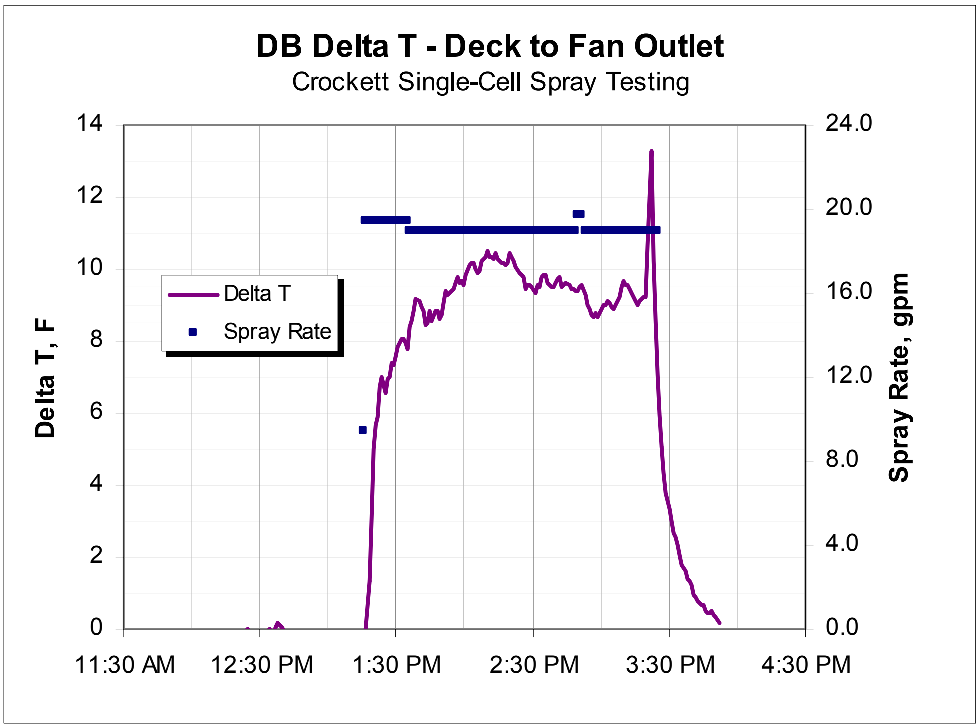


Figure 5-21
DB Delta T – Deck to Fan Outlet

ACE Time Constant

A decrease in the temperature of the water/glycol mixture exiting the cell and a decrease in the overall glycol loop temperature as the entire ACE cooling capacity is increased by the enhancement of the B-cell reflects the effects on enhanced cooling on the B-cell

The response time of the B-cell exit temperature is equal to the time required to flush out the cell and replace it with fresh fluid. Estimates of the internal volume of the finned tubes and headers suggest a cell capacity of 2000 to 2500 gallons. The measured flow rate through the cell is 1850 gallons giving a turnover time of approximately 1 to 1 ½ minutes.

The loop response time is determined by the turnover time of the entire auxiliary cooling loop. While no estimates of the total loop volume or coolant charge were available, it is reasonable for such systems to assume that the loop volume is 3 to 5 times that of the heat exchanger cells, which would the give a turnover time of no more than 5 to 10 minutes. Examination of plant readings (uncalibrated) of the overall ACE inlet and outlet temperatures, which track quite closely during transients with no apparent delay (see Figure 5-22), confirm this.

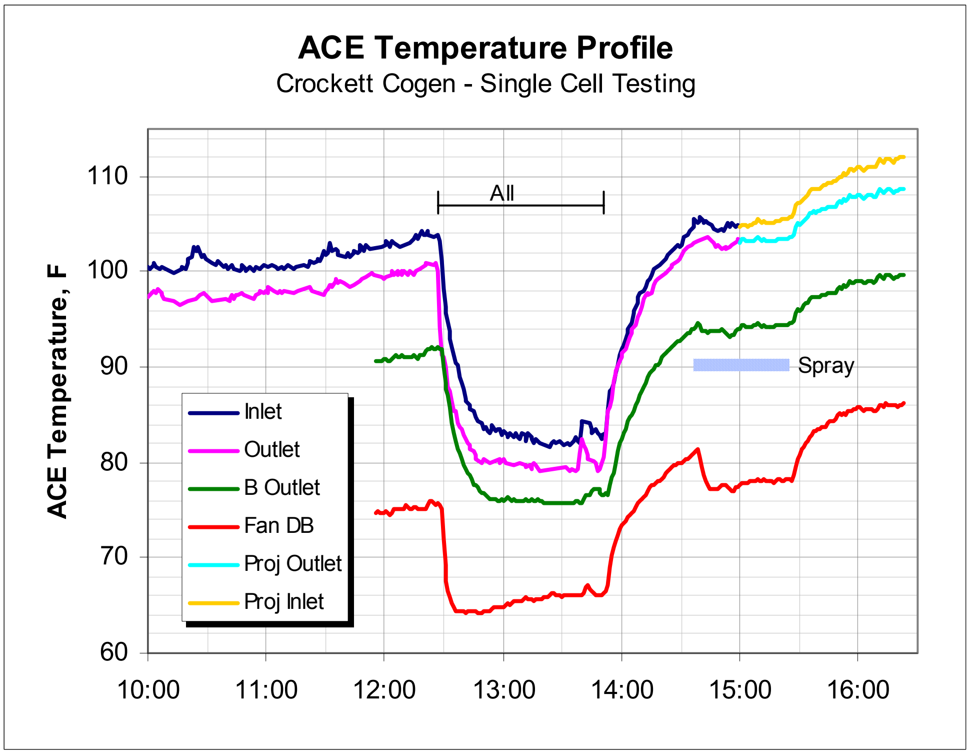


Figure 5-22
ACD Temperature Profile

Effect of Spray on Temperature Measurements

During the operation of the sprays, there is always some fraction of the water that does not evaporate. Unevaporated airborne droplets accumulate on all available solid surfaces, including the spray rack, the scaffolds, the fan, and shrouds as well as all the ACE support structure below and above the fan. Some of this water rains back, accumulates on the floor beneath the test cell, and eventually flows to the floor drains.

During the process, the dry-bulb temperature probes are susceptible to getting wet which introduces inaccuracies in the readings. Attempts to avoid this, which included shielding the fan outlet dry-bulb probes with metal covers and putting the deck-level and scaffold-mounted probes in aspirating psychrometers, were only partially successful. Figures 5-23 and 5-24 display the effect of the sprays on the temperature measurements. Dry operation with all fans running began at 1:45 P.M. and continued until the sprays were turned on at 2:12 P.M. The deck level dry-bulb measurements (Figure 5-23) were similar in spread and variability during both periods. The wind induced measurements in the north and east were somewhat higher than normal throughout the test period. When the sprays were turned on, the average appears to have lowered approximately 1° F.

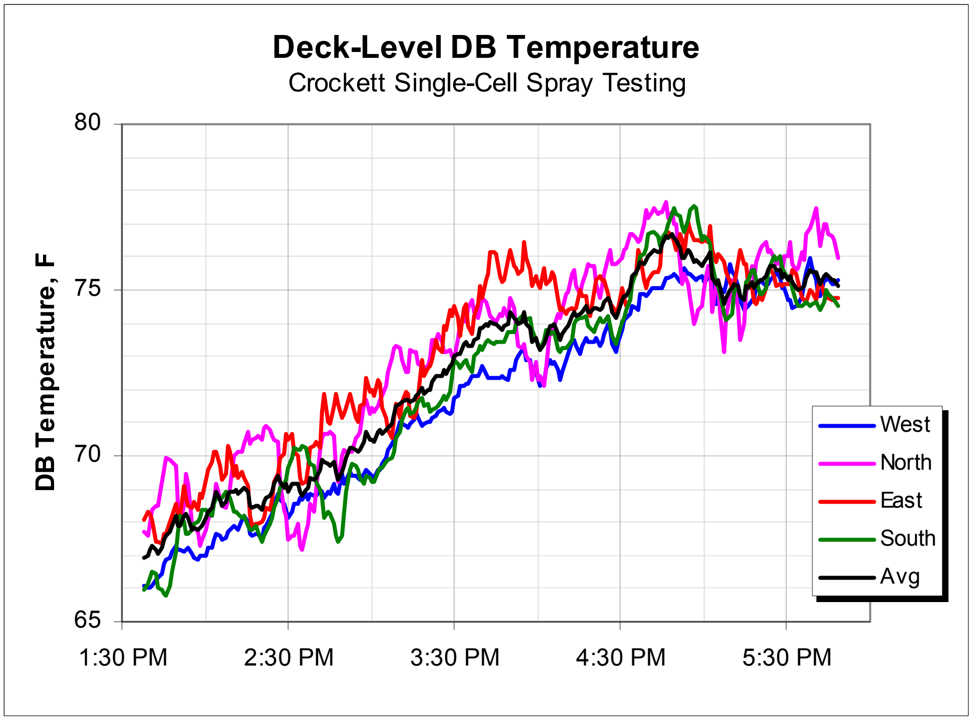


Figure 5-23
Deck-Level DB Temperature

The fan outlet measurements (Figure 5-24), however, showed a definite increase in spread when the sprays came on, particularly at two points in the northwest quadrant. With these two points excluded, the spread still changed from perhaps +/- ½° F during dry operation to +/- 1 to 1 ½° F during the spray period. The sprays seriously affected the scaffold-mounted probes(see Figure 5-25), with the north/top probe falling nearly to the wet-bulb temperature. (The apparently anomalous behavior of the south/floor measurement appears to be due to a combination of solar load from which the others were shaded and possible intrusion of the hot air stream from the turbine hall along the floor at the west end of the unit.) On other days with gustier wind and higher spray rates, the scaffold measurements could be completely separated over the range from dry-bulb to wet-bulb. Due to this extreme variability, researchers concluded that these measurements were not useful when the sprays were running and did not use them in any of the data analysis.

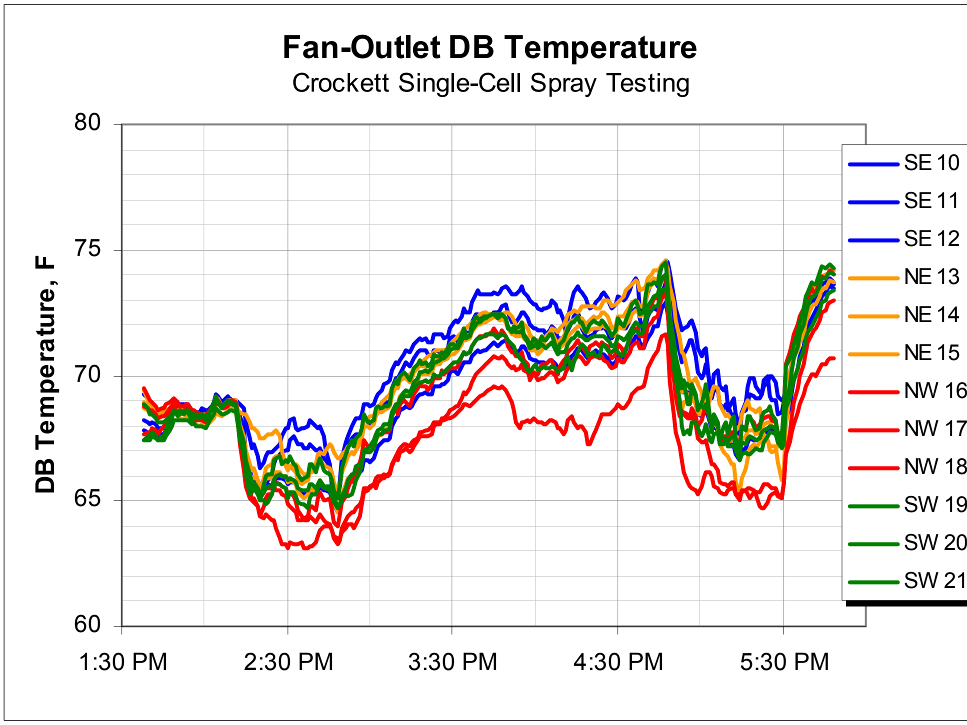


Figure 5-24
Fan-Outlet DB Temperature

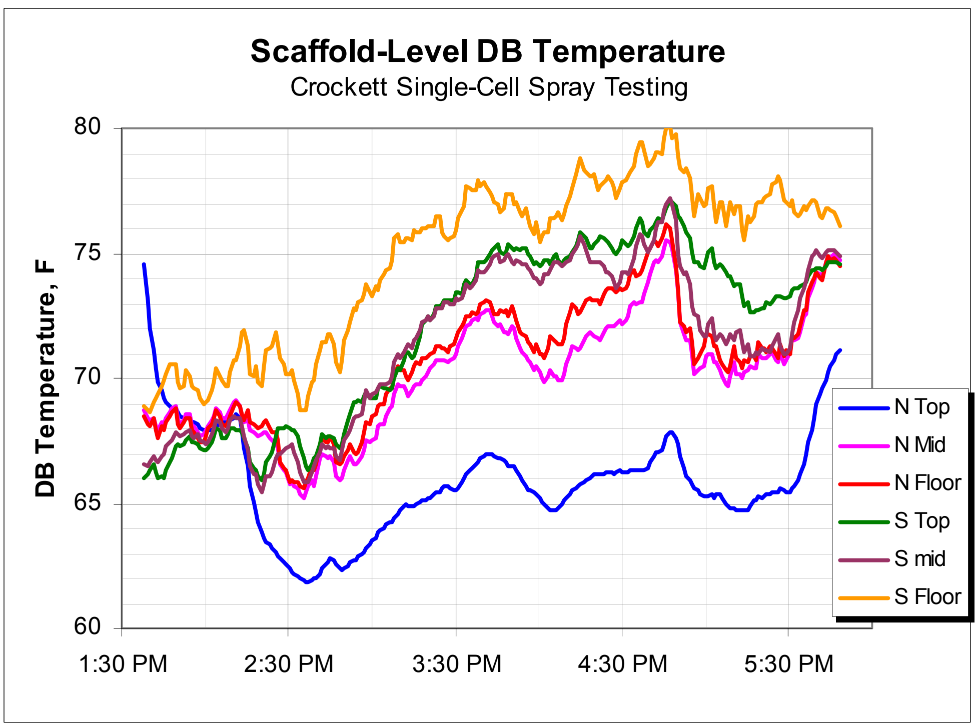


Figure 5-25
Scaffold-Level DB Temperature

As would be expected, the sprays had little effect on the wet-bulb measurements at the deck level and above the fan. The spread was typically +/- 1 to 1 ½° F at the deck level and +/- ½ to ¾° F above the fan. The averages were typically the same to within 1° F during both dry and spray operation, as shown in Figure 5-26.

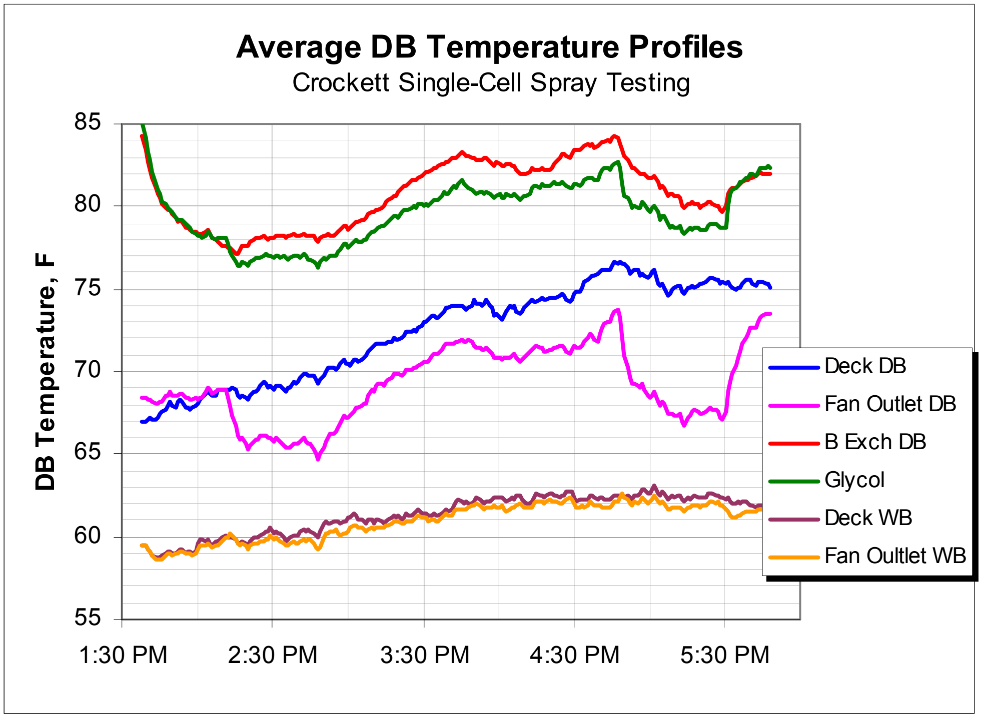


Figure 5-26
Average DB Temperature Profiles

Spray Accumulation on Solid Surfaces

The accumulation of sprayed water on the solid surfaces around the air-cooled exchanger causes another transient effect. Some of the spray deposits on the various components of the ACE including the spray rack, the scaffolding, the fan and its shroud, and the structural members above and below the fan. After depositing on these surfaces, the liquid leaves them through re-entrainment, evaporation, drainage, or rainback. A steady state accumulation is reached when the deposition rate, which is proportional to the spray rate, is balanced by the various removal mechanisms, which are proportional to the amount of liquid “held-up” on the surfaces.

An increase of the spray rate causes a period during which the deposition exceeds the removal rate until the accumulation reaches a new steady state value. During that time, spray is being effectively sequestered on the surfaces and hence is unavailable for evaporation and cooling. For this transient period, the cooling effect will be less than the steady state value for the new spray rate. Conversely, if the spray rate is decreased, the removal rate will exceed the deposition rate for a time, effectively providing excess liquid for evaporation and cooling, producing a cooling rate higher than the steady state value that it will be approaching.

Figure 5-27 shows this effect. Researchers changed the spray rate shortly after 3:00 P.M. and again at about 3:50 P.M. The cooling effect decreased from 10° F to 7.5° F in the first instance but did so over a period of about 12 minutes. After the second spray rate change, the cooling effect decreased from 8° F to 6° F, also in about 10 to 12 minutes. The further decrease from 6° F to between 5 and 5 ½° F was related to the decrease in ambient dry-bulb temperature between 4:00 and 4:30 P.M., as was the increase in cooling effect at constant spray rate between about 2:10 P.M. and 3:00 P.M.

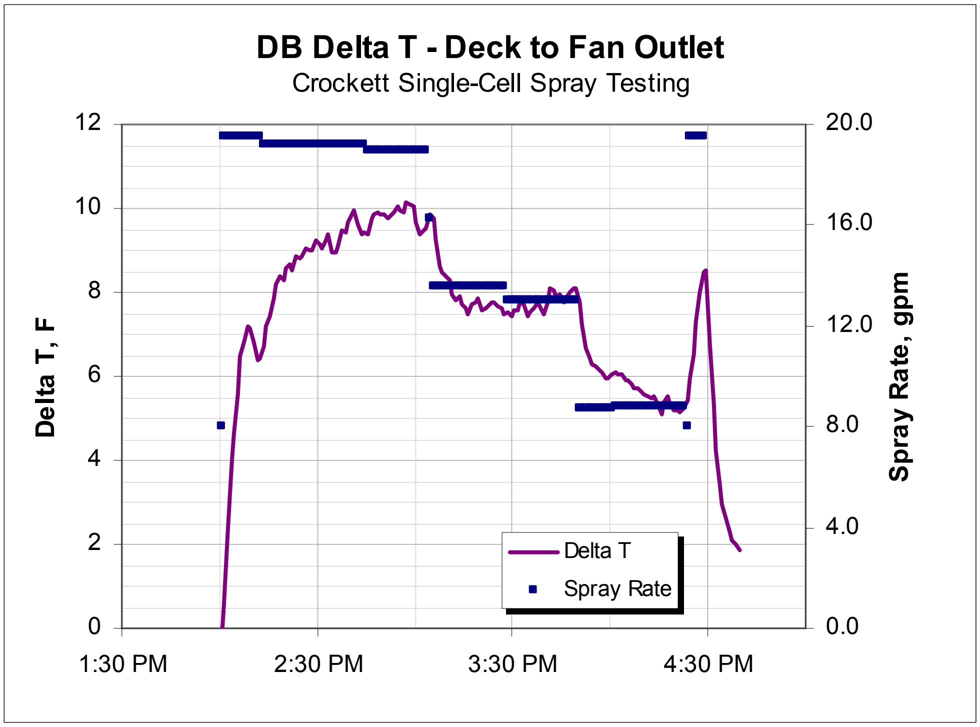


Figure 5-27
DB Delta T – Deck to Fan Outlet

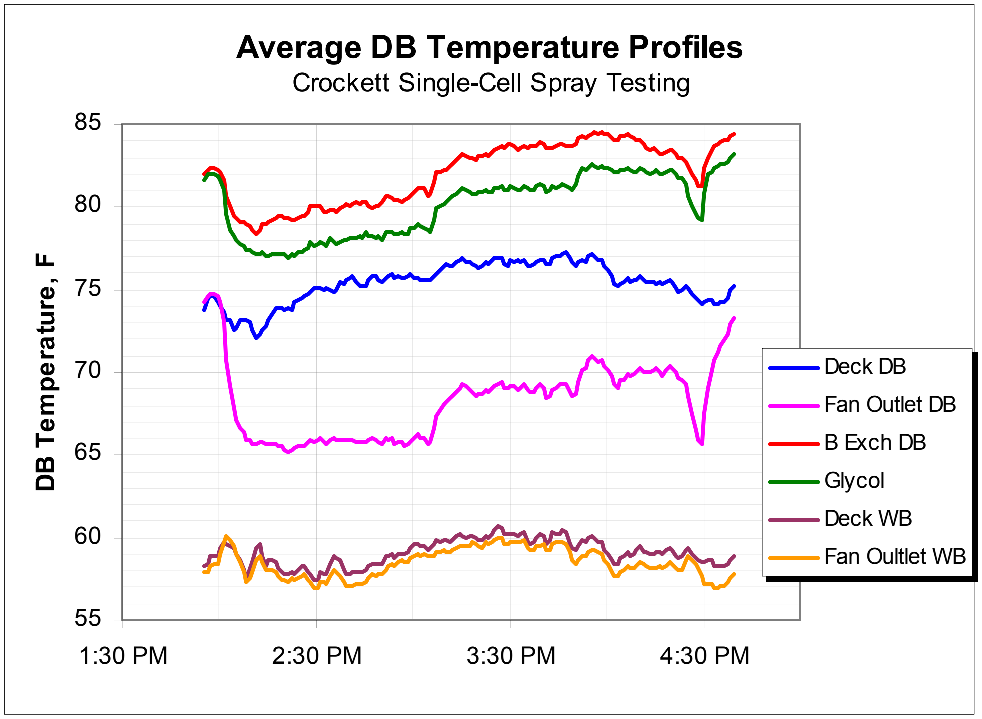


Figure 5-28
Average DB Temperature Profiles

6
SINGLE-CELL TEST RESULTS

Overview

Researchers conducted spray enhancement performance tests on Cell B of the air-cooled exchanger (ACE) at Crockett during 19 days between September 5 and November 9, 2002. Table 6-1 lists the test activity for each day of testing.

The objective of the single-cell test program was to determine the degree of performance enhancement that inlet spray cooling could achieve.

Table 6-1
Single-Cell Test Activities

DATE	ACTIVITY
Aug. 27-30	Set up instrumentation
Aug. 30	Air flow traverse and heat balance
Aug. 31	Instrument and data acquisition system checkout
Sept. 5	Dry operation to check effect of fan operation on inlet temperature; Short spray period to check spray flows and observe inlet spray patterns
Sept. 7	PJ-28 nozzles in place; rack at 1.5 ft. below fan First spray tests; one spray rate; one rack elevation
Sept. 10	Shielded probes above fan
Sept. 11	Lowered rack to 5 ft. below fan; Short spray tests; two flow rates; one rack elevation
Sept. 13	Tests at several spray rates by taking successive banks out of service; Same rack elevation as 9/11
Sept. 14	Lowered rack again to 8 ft. below fan; raised to 7 ft. because of winds Tests at several spray rates by taking successive banks out of service
Sept. 18	Ran ACE time response tests (fans on and off; sprays on and off).

Table 6-1 (continued)
Single-Cell Test Activities

DATE	ACTIVITY
Sept. 20	Demonstration for CEC visiting committee
Sept. 21	New, finer spray nozzles in banks 3 and 4; rack at 8 ft. below fan Ran with varying flow rates; ran with all new (finer) nozzles; all old (coarse) nozzles and mixture to observe wetting above fan and rainback
Sept. 26	All nozzles replaced with fine spray (PJ-20); rack at 8 ft. below fan Continuous run at one spray rate
Sept. 28	Rack at 9 ft. below fan; Tested at two spray rates
Oct. 1	Rack at 9 ft. below fan; Tested at two spray rates
Oct. 2	Tested at single flow rate; varied rack elevation to four different levels
Oct. 4	Installed swirl (pintleless) nozzles
Oct. 9	Rack at 7 ft. below fan; Tested 4 spray rates
Oct. 10	Tested at constant spray rate and four different rack levels
Oct. 12	Rack at 6.5 ft. below fan; Tested at several spray rates
Oct. 14	Rack at 6.5 ft. below fan; Tested at several spray rates
Oct. 24-25	Installed demister panels
Oct. 26	Operation with demister panels in place; Tested two spray rates and two rack elevations
Oct. 31	Air traverse with demister panels in place
Nov. 1	Rack at 10 ft. below fan; Tested several spray rates
Nov. 7	Removed spray panels
Nov. 9	Air traverse with no demister panels (rerun of Aug. 30 tests)
Nov. 13	Dismantled instrumentation and terminated single-cell test program

Cell Characterization

Before the beginning of the actual spray performance testing, researchers performed a heat balance on the cell to characterize the performance of the cell under dry conditions and to verify the adequacy of the instrumentation.

Heat Balance—Air Flow

Researchers took air velocity measurements at the exit of the finned tube bundles with a hand-held 9" propeller anemometer. They took sixty-four measurement points (sixteen on each of four tube bundles) on both the north and south faces. Each face has a superficial frontal area of 1632 square feet. Table 6-2 displays the results and Figures 6-1 and 6-2 plot them.

Table 6-2
Air Flow Data on Cell B (at exit of finned tube bundles)

	ACE Cell B - South Face				**ACE Cell B - North Face**			
	8/30/2001				8/30/2001			
Start	11:15	11:33	13:18	13:36	15:54	15:35	15:15	14:07
Stop	11:30	11:50	13:35	13:55	16:12	15:52	15:33	14:25
	Left to Right Direction (facing panels) >>>>				Left to Right Direction (facing panels) >>>>			
	Air Flow Rate, fpm				**Air Flow Rate, fpm**			
	A	**B**	**C**	**D**	**D**	**C**	**B**	**A**
1	370	500	460	640	640	380	400	410
2	530	450	600	720	760	610	430	580
3	440	440	520	550	590	490	390	540
4	480	430	500	660	690	530	470	480
5	600	470	450	560	590	280	400	510
6	510	490	500	560	660	660	490	520
7	500	410	500	540	580	200	330	430
8	520	430	510	580	730	470	470	450
9	380	430	530	550	760	170	300	350
10	520	340	460	430	710	510	460	400
11	510	520	480	750	730	410	400	420
12	530	390	700	430	490	580	580	570
13	410	530	690	590	430	730	540	430
14	500	330	770	390	290	640	870	570
15	260	390	610	260	260	550	630	420
16	350	380	480	280	250	300	640	340
Panel Average	**463**	**433**	**548**	**531**	**573**	**469**	**488**	**464**
		Average Flow Rate, fpm		**494**		**Average Flow Rate, fpm**		**498**

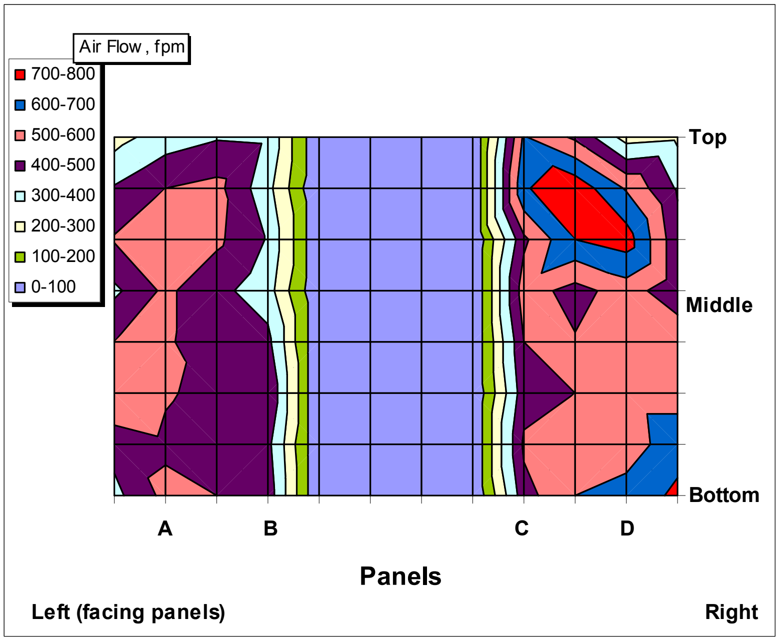


Figure 6-1
Air Velocity at Exit of South Face of Cell B

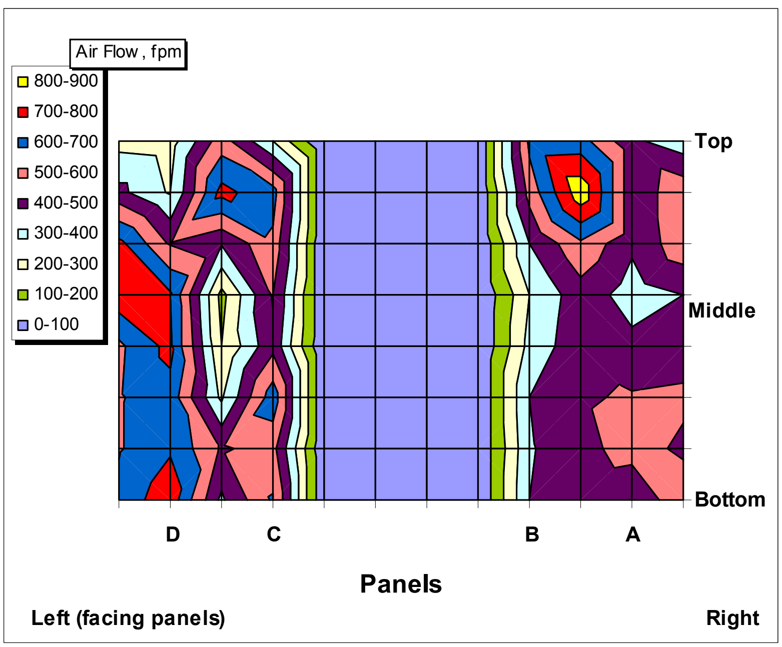


Figure 6-2
Air Velocity at Exit of North Face of Cell B

The following points are noteworthy:

- There is considerable variation from point to point:
  - On the south face: V_{max} = 770 fpm; V_{min} = 260 fpm
  - On the north face: V_{max} = 870 fpm; V_{min} = 170 fpm
- In spite of this variability, the total integrated flow is nearly the same of the two faces:
  - $Q_{South} = 8.06 \times 10^5$ ACFM
  - $Q_{North} = 8.13 \times 10^5$ ACFM
- The total flow is approximately 15% below that specified by the manufacturer.
- The short-term temporal variation at individual points is typically $\pm$ 2 to 10 % over 5 to 10 minutes.

Heat Balance—Coolant Flow

Researchers determined the coolant flow by velocity measurements, which they took in the two 10-inch inlet downcomers with an Ultrasonic velocimeter. The average velocity was 3.7 ft/sec. This corresponds to a coolant mass flow in Cell B of 9.16×10^5 lb/hr. Assuming equal flow in each of the three cells, the measured ACE flow is within 7 % of the manufacturer's specified flow rate.

Heat Balance—Heat Loads

Fluid properties (density and specific heat) for air and for a 10% propylene glycol solution were obtained from the ASHRAE Handbook {ASHRAE 1967 #140}. Figures 6-3 and 6-4 show air inlet and exit temperatures and the glycol exit temperature, monitored during the afternoon of August 31, 2001. Table 6-3 gives the average values during the period from 4:30 to 5:00 P.M. when the inlet conditions were most steady. Researchers obtained the glycol inlet temperature from the plant control room for the corresponding period.

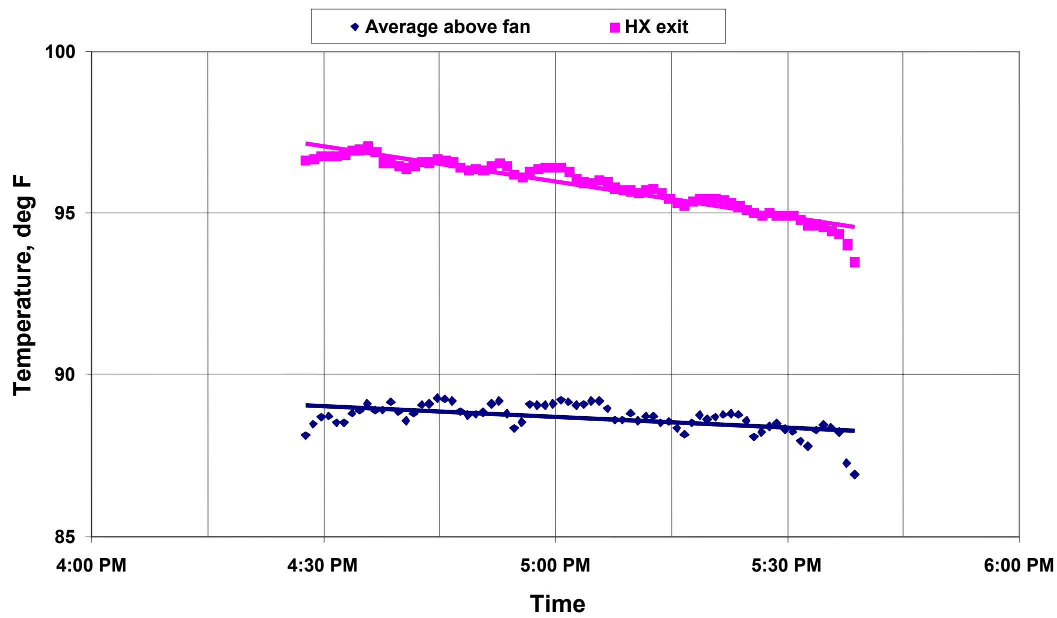


Figure 6-3
Air Temperature Rise Across Heat Exchanger

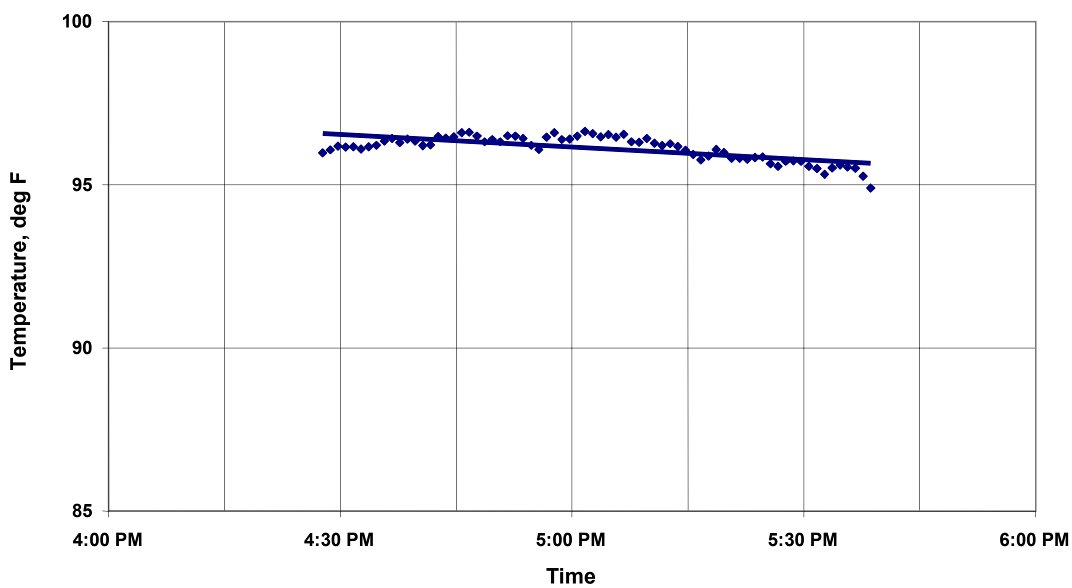


Figure 6-4
Cell B Glycol Exit Temperature

Table 6-3
Cell B Heat Balance (Based on averages from 4:30 to 5:00pm, August 31, 2001)

	Flow (lb/hr x 10^{-6})	Tin °F	Tex °F	Heat Load (Btu/hr x 10^{-6})
Coolant	.916	103.5	96.4	6.34
Air	3.48	88.9	96.5	6.35

The indicated heat balance is within 1%. While agreement this close is clearly fortuitous, it indicates that the instrumentation is adequate. In addition, the overall heat transfer capability, UA {Btu/hr – °F}, defined as heat load/mean temperature difference is within 8% of the value given in the manufacturer's specifications. This is consistent with the differences in the air and coolant flow rates and gives confidence that the measured performance during dry operation is reasonable and Cell B, as characterized, performs as expected from the specified design information.

Spray Enhancement Performance Tests

Researchers primarily determined performance of the spray enhancement approach by measuring the reduction in temperature of the heat exchanger inlet air. Other indicators included the following:

- The increase in air temperature rise across the heat exchanger, which is proportional to the heat load absorbed by Cell B
- The reduction in exit temperature of the water/glycol mixture

Researchers determined the amount of water evaporated in the inlet air stream from the change in specific humidity (lb moisture per lb of dry air) based on measurements of the airflow rate and the dry- and wet-bulb temperatures of the air below and above the sprays (see Chapter 4 for a description of the measurement locations). The amount of water evaporated, expressed as a fraction of the amount of water sprayed into the air stream, is a measure of the efficiency of water utilization as a cooling medium for the inlet air.

The controlled variables were spray rate, spray nozzle elevation, and droplet size distribution.

Spray Rate

Valving in and out one or more banks of nozzles varied the spray rate. The nozzles were arranged in eight independently valved branches. Each branch had seven connection points where one or more nozzles could be attached. Additional spray rate control was achieved by varying the supply pressure. However, changes in the pressure also produced a slight change in the droplet size distribution in the spray, making it difficult to separate the drop size effect from the spray rate effect. Therefore, all tests were run at a supply pressure of 300 psig except for a few runs when the pressure had to be reduced to 200 psig to prevent rupture of the supply hoses. This was required when plant operations caused the spray water to be delivered to the test apparatus at temperatures above 120° F.

Nozzle Location

Raising or lowering the spray rack (see Figure 4-1), which was suspended below the fan on cables run over pulleys hung from the ACC support structure, varied the spray nozzle location. Testers could raise the rack to the bottom of the fan shroud, approximately 1 ½ feet below the leading edges of the fan blades or lower it to the floor, about 30 feet below the fans. They conducted most of the testing at levels between 2 and 10 feet below the fans.

The intent of varying the rack elevation was to change the residence time of the droplets in the air stream before they entered the heat exchanger. However, if researchers lowered the rack more than about 10 feet below the fans, much of the spray was blown out from under the B-cell before it could be entrained. It then either fell to the deck or was entrained by one of the condenser cell fans. This made it impossible to establish a quantitative relationship between the spray rate and the cooling effect in Cell-B, so such operating conditions were avoided.

Drop Size Distribution

Changing the nozzles varied the drop size distribution. Tests used three nozzles based on their availability and performance in laboratory pretesting (see Chapter 4). Table 5-1 lists the nozzles and their operating characteristics. The nozzles with finer sprays also had lower flow rates per nozzle. Therefore, more of the finer nozzles were installed so that researchers could compare performance at equal spray rates with different drop size distributions.

Equipment and plant operations set the airflow, coolant flow, and heat load on the air-cooled exchanger. The air and coolant flow rates were assumed constant throughout the tests. In the case of the air flow there was likely to have been some variation due to wind effects, but it could neither be controlled nor monitored. Plant operations set the heat load on the cell but it varied little during most periods.

Additional uncontrolled variables were ambient atmospheric conditions including dry- and wet-bulb temperatures and wind speed and direction. Chapter 5 discussed the effect of rapid changes in wind conditions . Researchers monitored the variation in ambient temperatures in all tests and, while important in determining the evaporation rate and cooling effect, it typically varied slowly enough so that quasi-steady conditions could be assumed.

Response to Spray Enhancement—A Typical Run

Figure 6-5 displays the average temperature profiles during a long-term, quasi-steady state run on September 26, 2002 from 12:00 noon to 3:15 P.M. Conditions for this run were as follows:

- Nozzle: Impingement pin type; Model # EC-IP28 (90° cone angle; 0.36 gpm @ 300 psig; D(3,2) ~ 35μ) {D(3,2) = Sauter mean dia.}[1]
- Rack elevation: 8 ft
- Spray rate: 19 gpm

The ambient dry-bulb temperature increased steadily from 70° F to 80° F between noon and 2:15 P.M. and then remained level between 78° F and 80° F until 3:15 P.M. The time between 1:15 to 3:10 P.M. represented the most stable operating period. The abrupt temperature increase in heat exchanger exit temperatures (BExch DB and "Glycol") at 3:10 was due to a change in plant operation resulting in a 50% increase on the heat load to Cell B as indicated by the air temperature rise across the exchanger increasing from 12° F to 18° F. The apparent spike in the ambient dry-bulb temperature (Deck DB) resulted from hot spray from a ruptured hose hitting the deck level temperature probe on the north side and may be ignored.

The temperature profiles shown in Figure 6-5 indicate the following:

1. Before 1:15 P.M., the unit was running with all fans on and all cells dry. The general increase in temperature at all points tracks the increasing ambient (Deck DB) temperature.

2. During dry operation, the air temperature rise across the heat exchanger averages about 6.7° F after a quasi-steady state is reached at about 12:30 P.M.

3. With the onset of spraying at 1:15 P.M., the inlet temperature to the heat exchanger (Fan Outlet DB) drops by approximately 6.5° F from 75.5° F to 69° F.

4. The ambient wet bulb at temperature at that time was about 62° F, giving a wet-bulb depression of 13.5° F (76.5–62° F). Therefore, the initial cooling effect of 6.5° F is about 48% of the maximum cooling available.

5. The cooling effect varies from 6.5 to 10.5° F during the test period; the wet-bulb depression varies from 13.5 to 16° F.

6. The temperature rise of the air across the heat exchanger (BExch DB – Fan Outlet DB) increases from 6.7° F to 12° F, indicating a near doubling (~ x1.8) of the heat load carried by the enhanced cell.

7. The relative humidity of the inlet air increases from approximately 43 to 73% (see Figure 6-6).

[1] The Sauter mean diameter (D(3,2) is defined at that diameter for which, if all the droplets were the same size, the spray would have the same surface area.

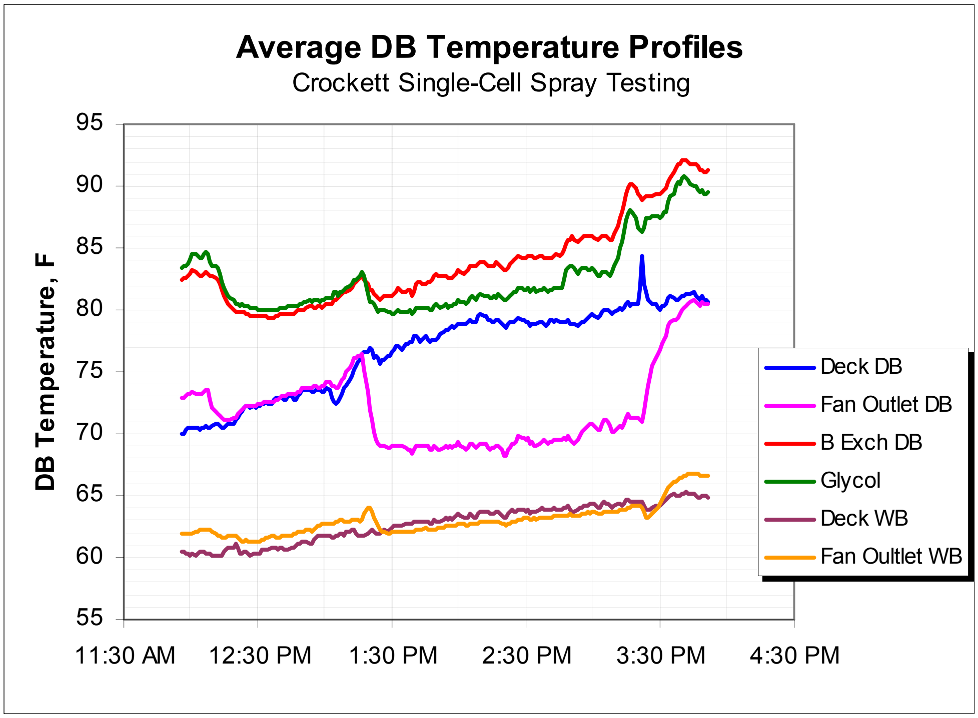


Figure 6-5
Long Term Run of Sept. 26

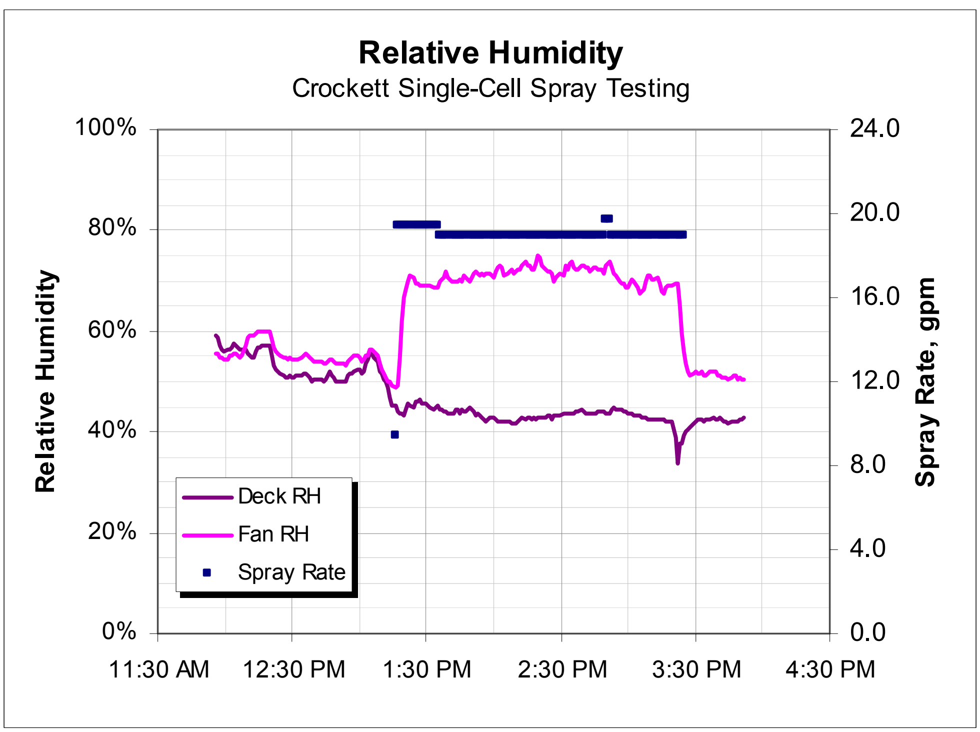


Figure 6-6
Relative Humidity Increase

8. During the period of spray operation from 1:15 to 3:15 P.M., the wet-bulb depression and the inlet cooling effect varied as Figure 6-7 shows. The heat exchanger inlet temperature (Fan Outlet DB) remained nearly constant between 69 and 71° F. The cooling effect varied from 50 to 65% of the wet-bulb depression over the time period as the ambient temperature changed.

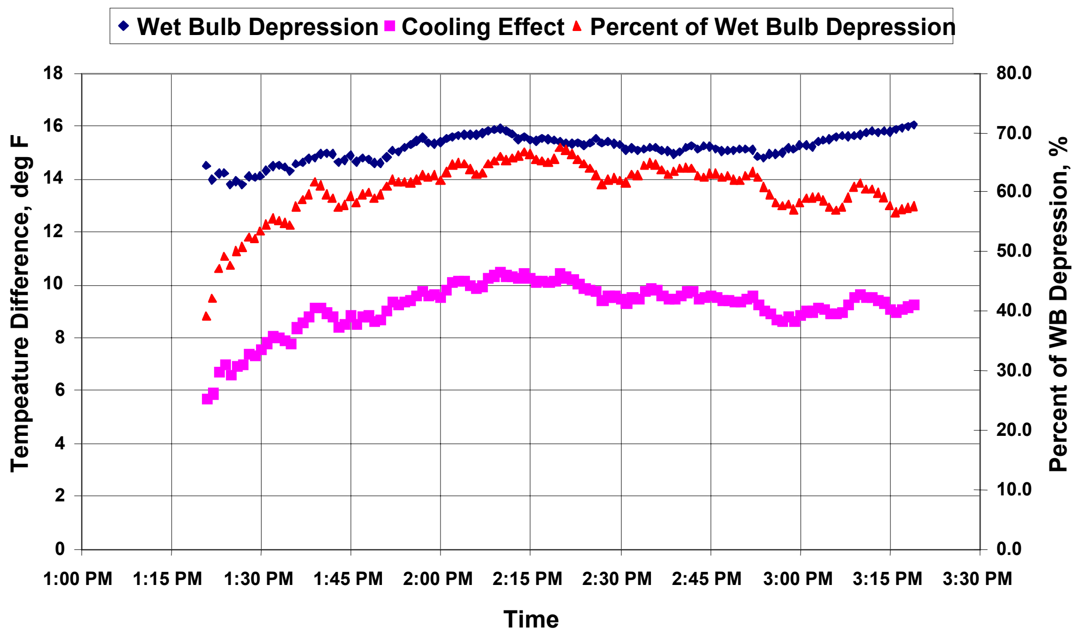


Figure 6-7
Spray Cooling Performance--9/26/02

9. Researchers calculate the evaporation rate from the change in specific humidity (lb of moisture per lb of dry air, determined from the measured dry- and wet-bulb temperatures above and below the spray injection point) times the airflow rate. Figure 6-8 shows the result as both absolute evaporation rate and fraction of the spray rate. It is apparent that 60 to 70% of the sprayed water is evaporated and contributes to the cooling effect. At constant spray rate, drop size distribution, and residence time (rack elevation), it varies primarily with the wet-bulb depression, which is the effective driving force for evaporation. As the discussion of the laboratory test results in Chapter 4 noted, the difference in specific humidities is very sensitive to slight errors in the measurement of the wet-bulb temperatures. To minimize the appearance of this sensitivity, 15-minute rolling averages are displayed for the evaporation rate and utilization efficiency rather than minute-by-minute data points.

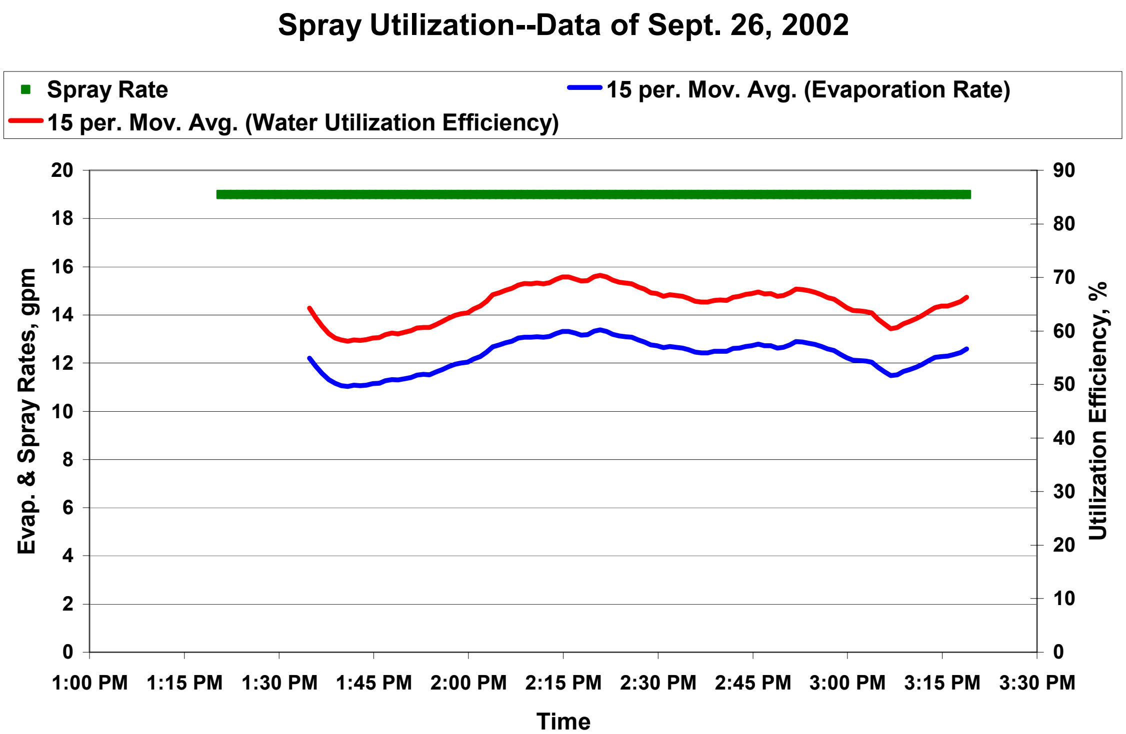


Figure 6-8
Spray Utilization – Data of Sept. 26, 2002

Effects of Design and Operating Variables

Method of Data Presentation

Researchers plotted the temperature profiles and performance runs presented in Figures 6-5 through 6-8 against clock time. During the test period the controlled variables of spray flow rate, spray rack elevation, nozzle type, and spray supply pressure were held constant. The observed variations in cooling effect, evaporation rate, and operating temperature levels are attributable to variations in air inlet (ambient) dry- and wet-bulb temperatures and wind conditions.

Variations in ambient temperatures change the driving force for droplet evaporation and set the maximum cooling effect that can be obtained. The maximum cooling effect from adiabatic evaporation is the wet-bulb depression (WBD), defined as the difference between the ambient dry- and wet-bulb temperatures. Figures 6-9 and 6-10 present the cooling effect against the ambient dry-bulb temperature (Figure 6-9) and against wet bulb depression (Figure 6-10). The cooling effect is defined as the difference between the inlet (ambient) air temperature and the air temperature measured above the fan, which is taken as approximately equal to the temperature of the air entering the finned tube bundles of the air-cooled heat exchanger.

Figure 6-9 shows a general increase in cooling effect with ambient temperature and a leveling off and some increased scatter for ambient temperatures between 79 and 80.5° F. This behavior is consistent with the time history of the ambient conditions displayed in Figure 6-5, in which the ambient dry-bulb increased relatively smoothly from 75 to 80° F, then leveled off and varied narrowly between 79 and 80.5° F for the reminder of the run. During that time, the wet-bulb continued to rise with a resultant reduction in the wet-bulb depression. This accounts for the cluster of data points with somewhat increased scatter around 79° F ambient dry-bulb.

Figure 6-10 displays the same cooling effect data plotted against wet-bulb depression exhibiting a smoother relationship with less scatter. Figure 6-10 also shows the maximum cooling that could be achieved if the entire spray flow of 19 gpm were completely evaporated. At the highest wet-bulb depression of 16° F, approximately 80 to 85% of the maximum cooling effect was obtained.

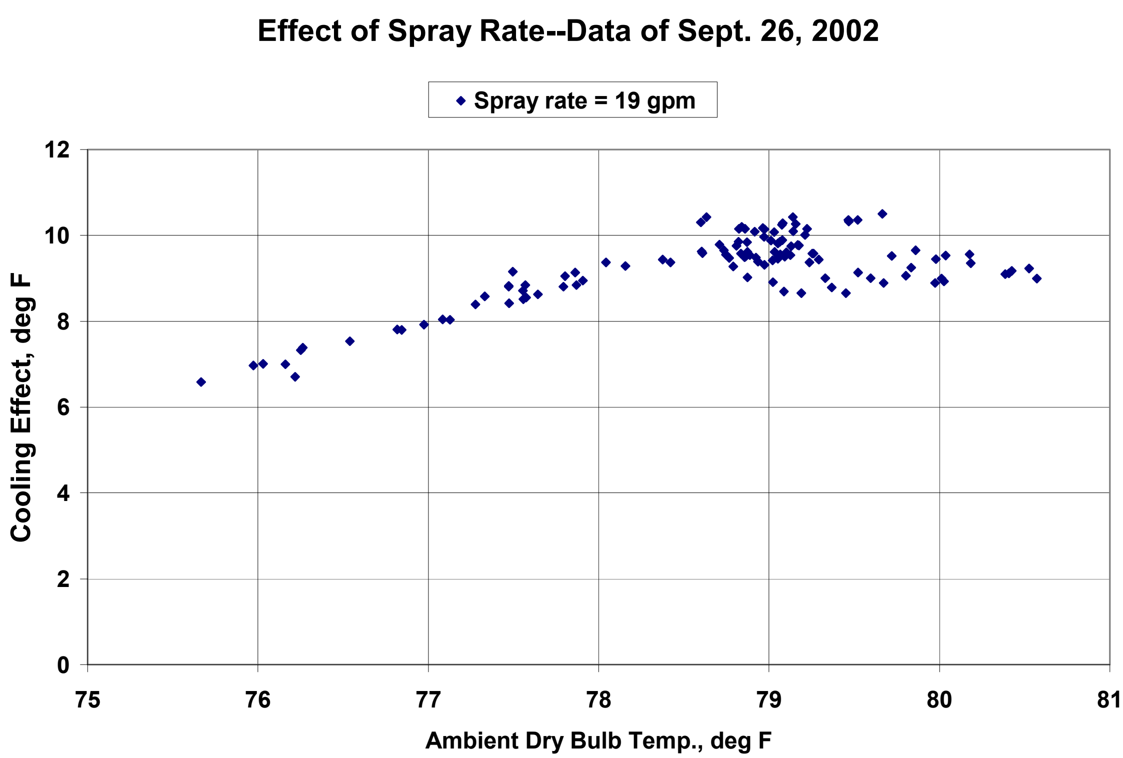


Figure 6-9
Effect of Spray Rate – Data of Sept. 26, 2002

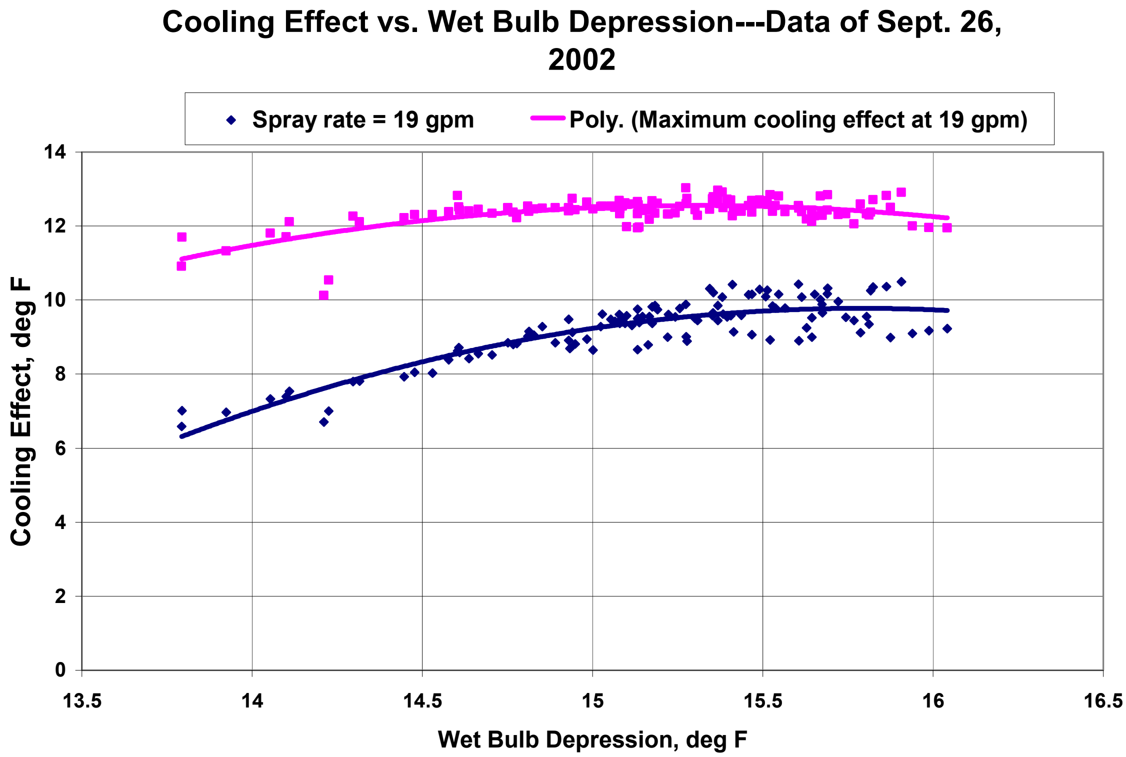


Figure 6-10
Cooling Effect vs. Wet Bulb Depression – Data of Sept. 26, 2002

Effect of Spray Flow Rate

Spray flow rate has a major influence on the cooling effect, the fractional evaporation, and the degree of unevaporated water falling back from the spray region.

Figure 6-11 illustrates the effects, displaying system performance at three flow rates: 19, 13, and 8.6 gpm. Researchers turned on the sprays at 1:00pm and initially set them at 19 gpm. They reduced the flow twice during the test period—at 3:05 P.M. to 13.6 gpm and again at 3:50 P.M. to 8.6 gpm. In between changes they held the flow steady. Just before the end of the test (4:25 P.M.), they increased the flow back to 19 gpm for about 5 minutes.

As the spray flow successively reduces, a general stepwise pattern of increasing air inlet, air exit, and coolant exit temperature is apparent. The abrupt drop in temperature just before the termination of the run is the effect of restoring the spray flow to the initial high rate of 19 gpm. Variability during the periods of nominally steady state operation is attributable in part to the continuous variations in the ambient dry- and wet-bulb temperatures during the course of the test.

Figure 6-12 shows the cooling effect as a function of ambient dry bulb temperature. Several features of this plot are noteworthy.

1. The data show a clear effect of spray rate with the higher spray rates providing a greater cooling effect at the same ambient temperature. (This same effect is seen against wet-bulb depression as well in Figure 6-14.)

2. The time history of the data points can be discussed by referencing back to Figure 6-11. The test proceeded from the low, left-hand end of the series of 19 gpm points (blue diamonds) upward and to the right. The transition from the 19 gpm to the 13 gpm spray rates is seen in a series of five points (purple squares) descending to the right, from performance consistent with the higher spray rate to the cluster of points at the lower cooling effect. During the time when the spray rate was set at 13.6 gpm, the ambient temperature leveled off (see Figure 6-11).

   The series of four or five transitional points (green triangles) shows the second spray rate reduction, again descending to the right from the 13 gpm cluster to the new level of the 8.6 gpm data. At approximately the time of the second reduction in spray rate, the ambient temperature began to decline and the test points descend to the left.

3. When researchers increased the spray rate back to the 19 gpm level, the operating points proceed upward to the left (red triangles) back to a cooling effect consistent with the earlier data at 19 gpm. Chapter 5 discussed this transitional behavior qualitatively as part of the discussion of transient effects on the measurements and attributed it to the accumulation and depletion of excess liquid on the solid surfaces of the ACE, which adds to or diminishes the spray available for evaporation following changes in spray rate.

4. Figure 6-13 displays the same data with the transitional points deleted to show the separation with flow rate more clearly. Note the two (red triangle) points showing that the performance returned to the expected level when the spray rate was returned to its earlier value.

5. Figure 6-14 shows the cooling effect as a function of wet-bulb depression along with the maximum cooling effect that could be achieved with each of the three flow rates. At the higher flow rate, the measured cooling effect is again 80 to 85% of the maximum, as it was during the September 26 runs (see Figure 6-7). At the lower flow rates, the cooling effect reached almost 100% of the maximum achievable amount indicating that essentially all the spray was evaporated.

6. Finally, Figure 6-15 shows a comparison of the performance at similar flow rates and rack elevation and using the same nozzles on two different days, presented as cooling effect vs. wet bulb depression. The cooling effect on September 26 was higher by as much was 2° F than at similar conditions on September 28. This appears to be attributable to the higher ambient temperature on September 26. Even though the wet- bulb depression (the driving force for evaporation) was the same, the higher temperature level and resulting higher vapor pressure of the liquid at the surface of the spray droplets makes the evaporation proceed more rapidly. At conditions where the evaporation is less than complete, the faster rate can produce a higher cooling effect in the finite time available for evaporation.

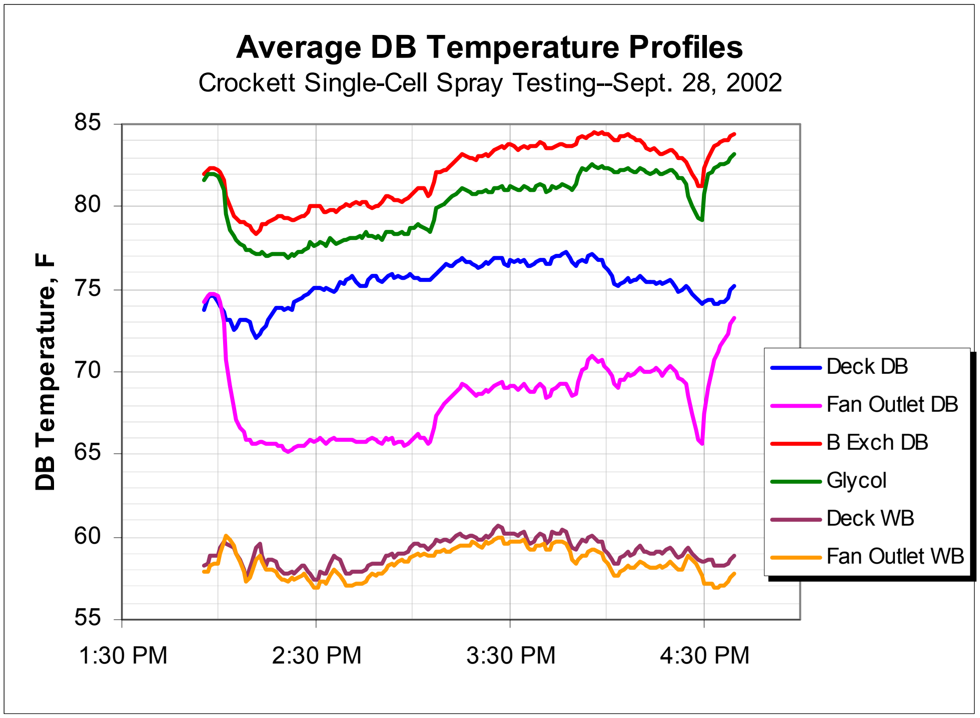


Figure 6-11
Run with Flow Rate Variation

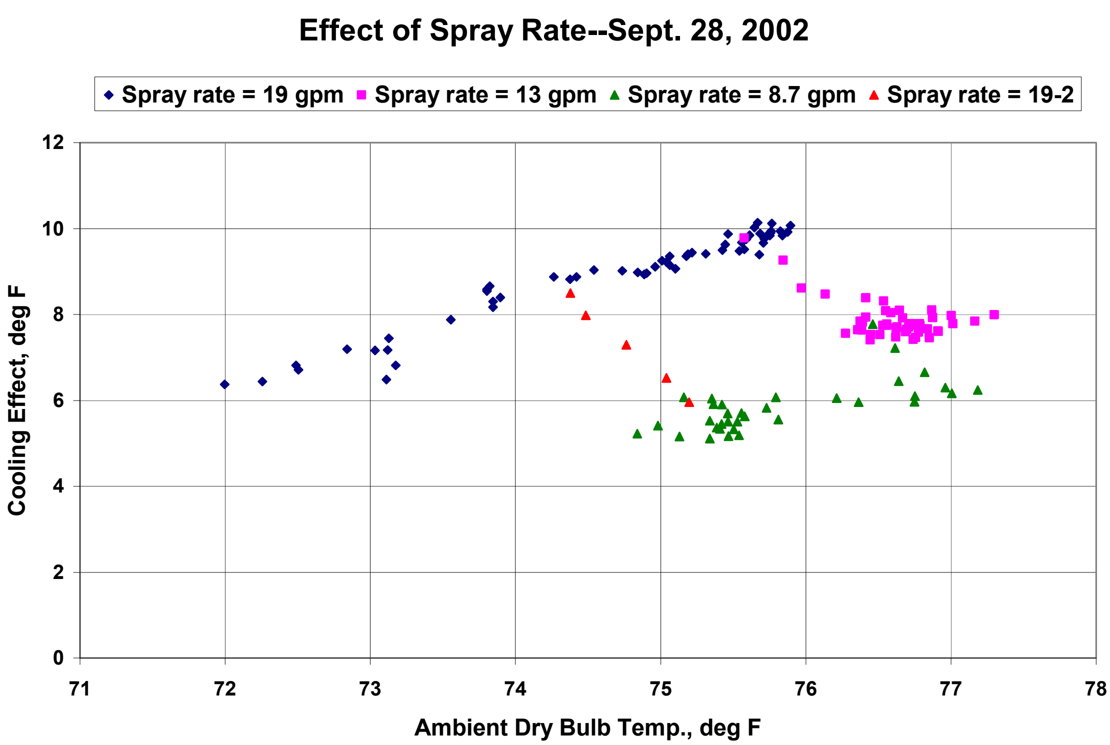


Figure 6-12
Effect of Sprays (with Transitions)

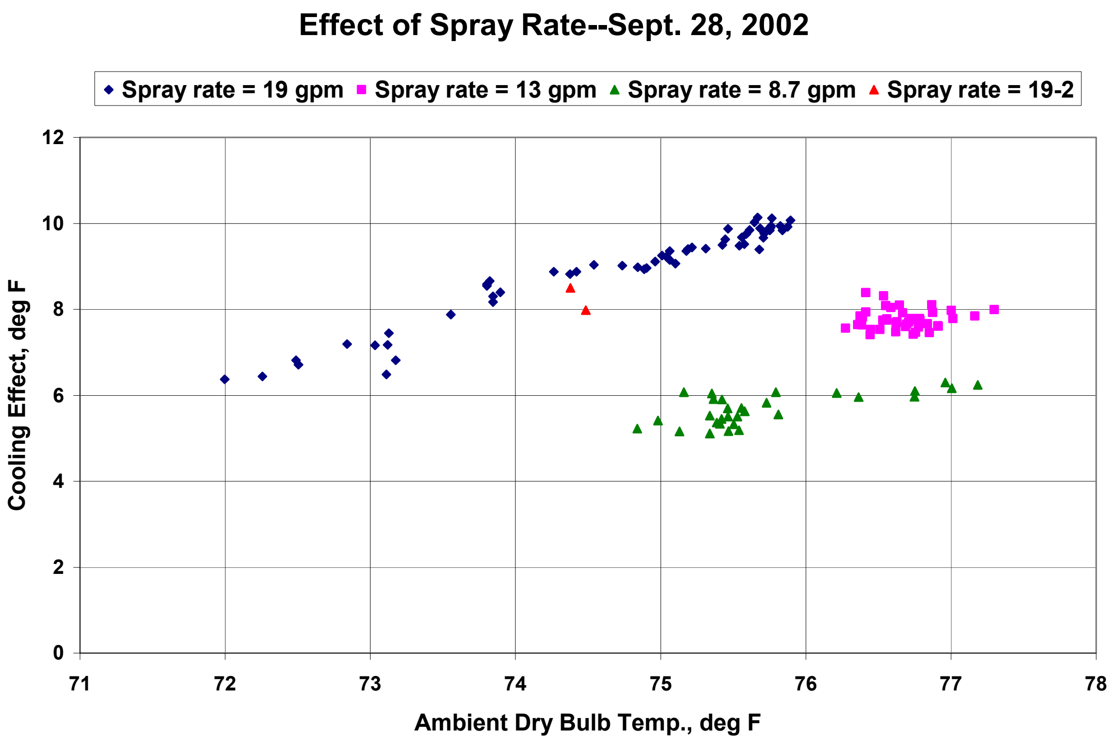


Figure 6-13
Effect of Spray Rate (w/o Transitions)

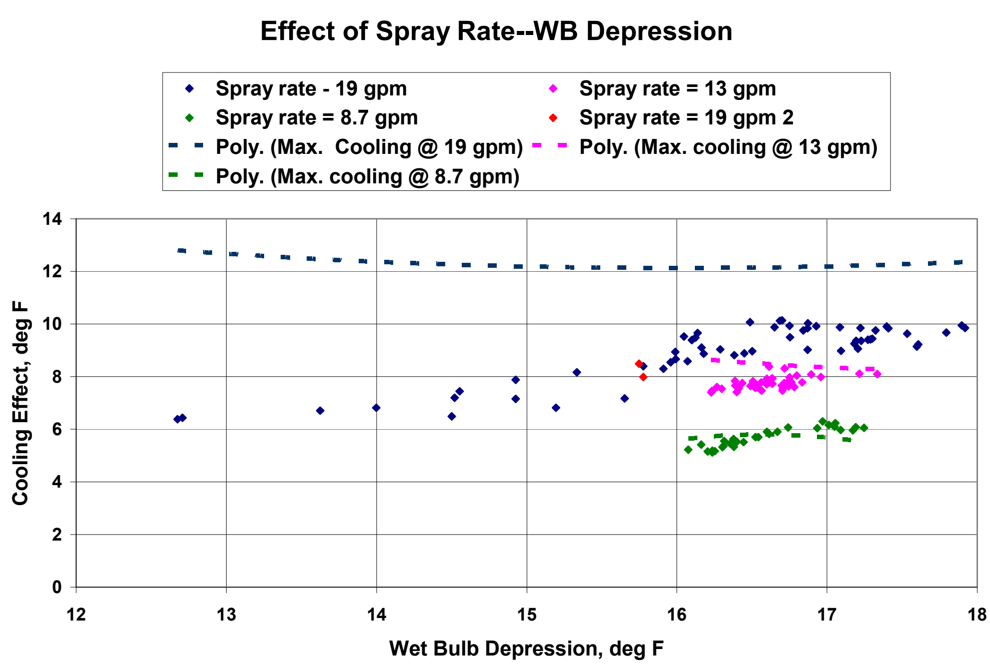


Figure 6-14
Cooling Effect vs. Wet Bulb Depression

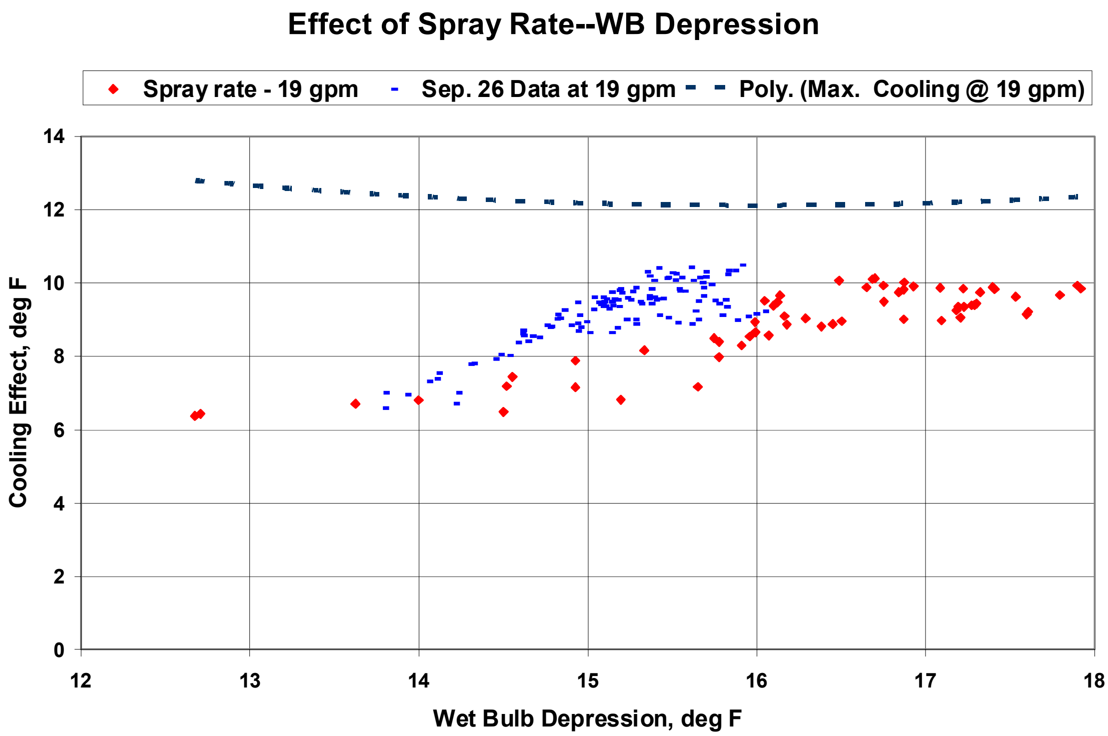


Figure 6-15
Comparison of Sept. 26 with Sept. 28 at 19 gpm

Effect of Nozzle Location

Varying the location at which the spray enters the inlet air stream will change the residence time during which droplets can evaporate and contribute to the cooling effect. Droplets that contact solid surfaces (such as ACC structural members and fan blades) before evaporating typically agglomerate, drain, and fall back into the air stream as new droplets too large to be re-entrained or contribute to any further cooling. Unevaporated droplets that enter the finned tube banks will normally evaporate on or near the fins and will enhance the cooling effect. However, they may also cause long term harm to the heat exchanger surfaces such as fouling, scaling, and corrosion.

Researchers mounted the nozzles on a rack that could be raised and lowered from approximately 1.5 to 9 feet below the inlet to the fan shroud or about 3.5 to 11 feet below the leading edges of the fan blades in order to provide some ability to investigate the effect of residence time. They expected that a location closer to the fan would provide a shorter residence time and hence less evaporation and cooling.

Two separate test runs varied rack elevation systematically while holding spray rate constant. On October 2, the nozzles were set at three levels of 4.3, 6.5, and 9 feet below the bottom of the fan shroud. The spray flow rate was set to 14 gpm. The nozzles installed for those runs were the finer of the two pintle nozzles (PJ20; impingement pin; mean droplet diameter 35µ). Figure 6-16 plots cooling effect data at each of the three levels against wet-bulb depression. There is no discernible effect of the elevation changes within the scatter of the data (+/- ½° F) during the test period. However, conditions during this test period were highly variable. As Figure 6-17 shows, the variations in cooling effect were significant during the periods of supposed steady state operation at constant rack elevation. Figures 6-18 and 6-19 indicate that significant changes in the wind conditions and the inlet wet-bulb temperature took place at 3:00 P.M. and 4:00 pm, which were just the times that the rack elevations were changed. The effects of the external weather conditions might well have obscured any obvious effect of the change in elevation and residence time.

Therefore, researchers conducted a second set of tests to confirm the results under less variable conditions on October 14. In this run, the rack elevations were higher (from 2 to 6 ½ feet from the shroud inlet), the spray rate was lower (~ 12 gpm), and different nozzles were in place (internal swirl nozzles; WDB14-90 with a comparable mean diameter of 38µ). Figure 6-20 plots the cooling effect data against wet-bulb depression, again showing no distinction among the results at the different elevations.

There may be many reasons for the absence of an observable effect:

1. The change in residence time from a 6-foot change in rack elevation is not large. The superficial air velocity at the fan inlet is approximately 20 feet per second. Therefore, a 6-foot change corresponds to only a 0.3 second change in residence time. The maximum residence time from the average spray nozzle location to the inlet of the finned tube banks is about 2 to 3 seconds so a change in residence of time of only about 10 to 15% is to be expected.

2. Even under relatively stable conditions, such as was the case on October 10, the air movement on the deck beneath the fans is moderate to strong and erratic, resulting in downdrafts that entrain and swirl much of the spray to levels below the nozzles before it is finally ingested into the fan inlet. As a result, these secondary flow patterns determined the residence time for much of the spray and the residence time was both larger and less predictable than would be calculated using average superficial velocity and the injection point. In addition, at the lower elevations, a greater fraction of the spray is blown out from under Cell B to be entrained by other cells, hence reducing the cooling effect on Cell B.

The net result is very difficult to estimate and obscures any expected systematic effect of nozzle elevation. This is particularly true when attempting to compare results taken on different days, where even small differences in more dominant variables, such as spray rate and ambient conditions, overwhelm the observable effect of nozzle height.

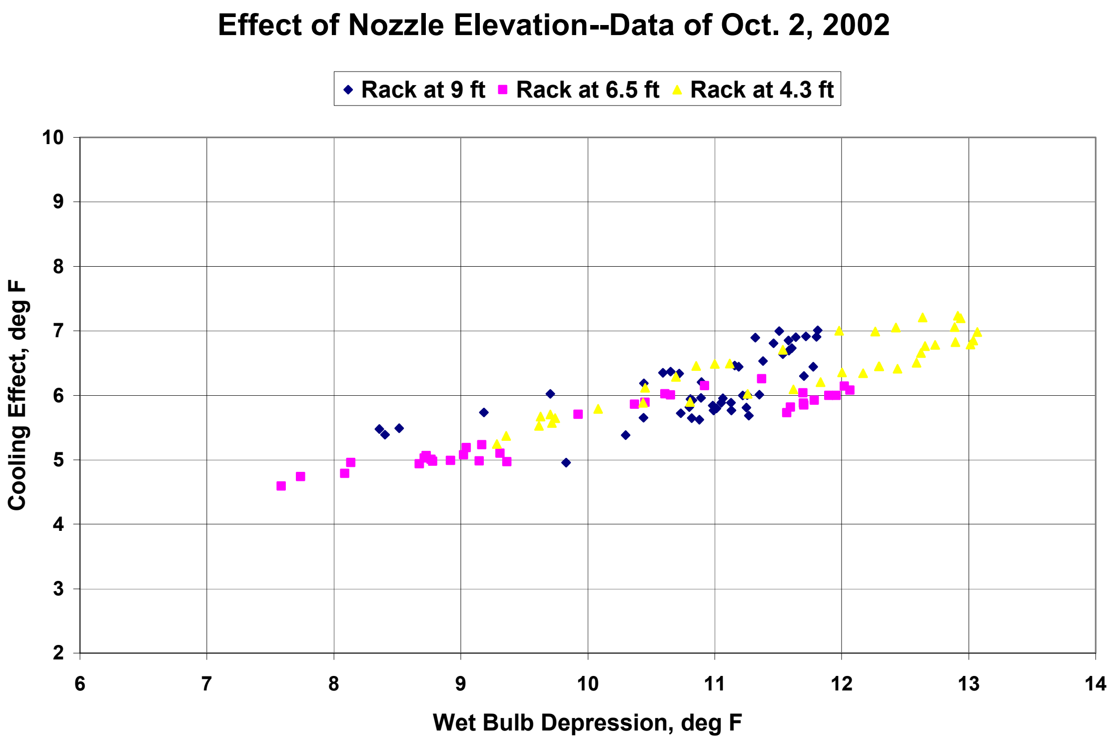


Figure 6-16
Effect of Nozzle Elevation – Data of Oct. 2, 2002

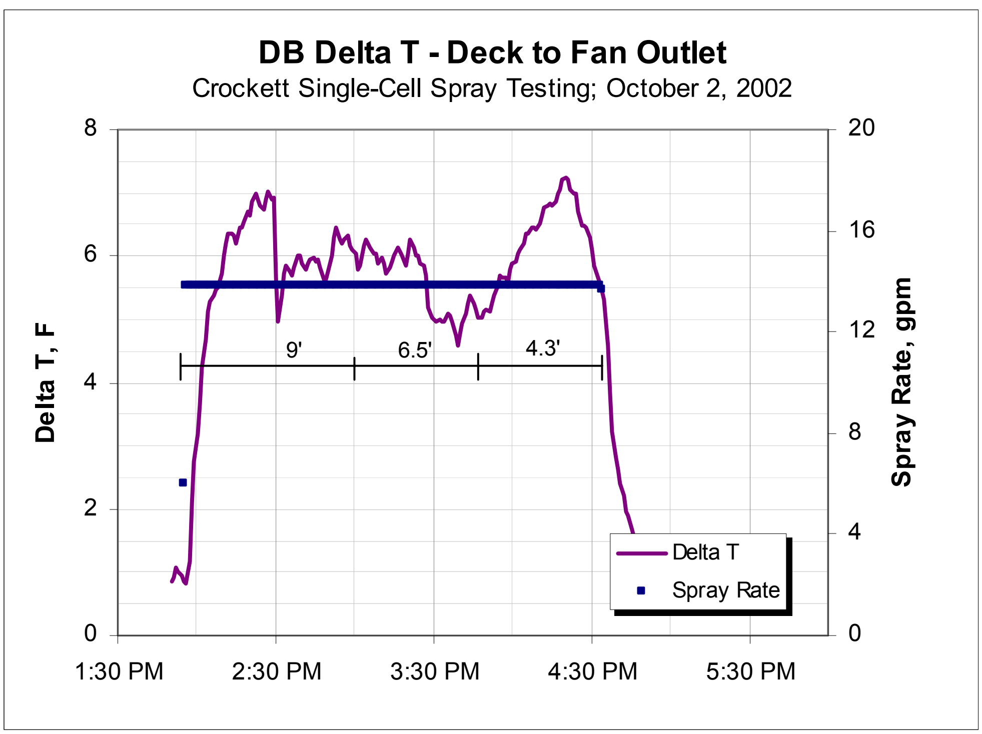


Figure 6-17
Effect of Nozzle Elevation

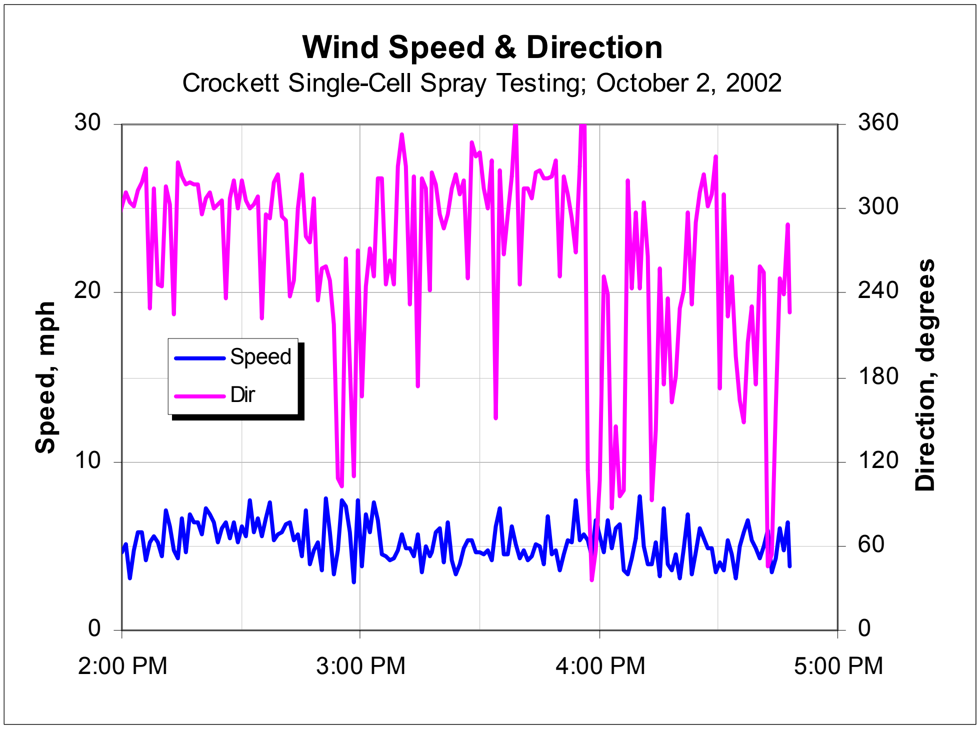


Figure 6-18
Wind Speed & Direction

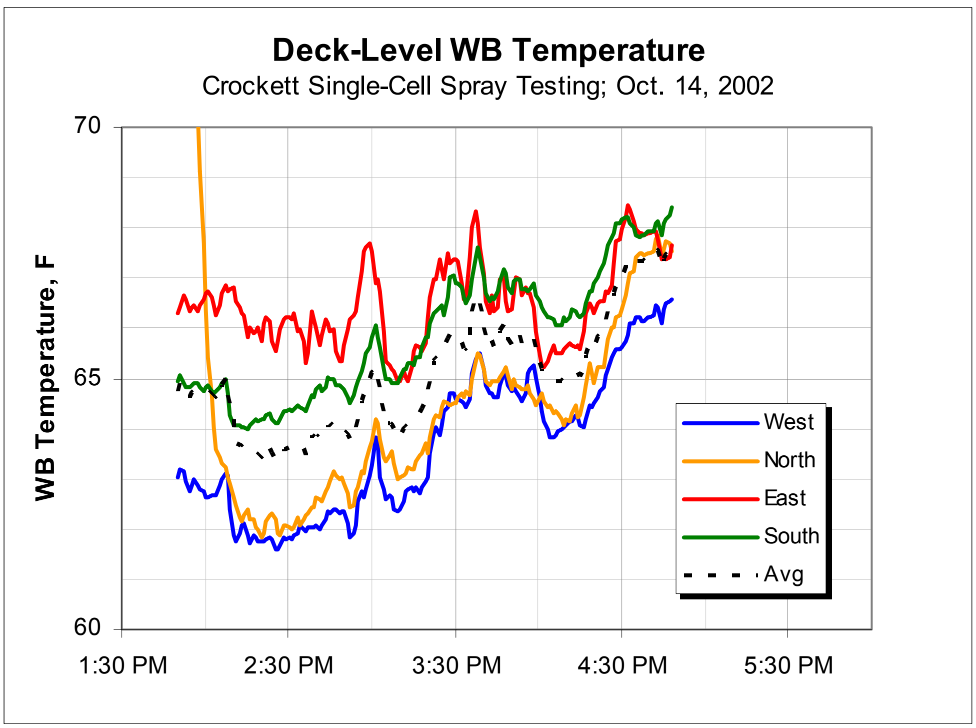


Figure 6-19
Deck-Level WB Temperature

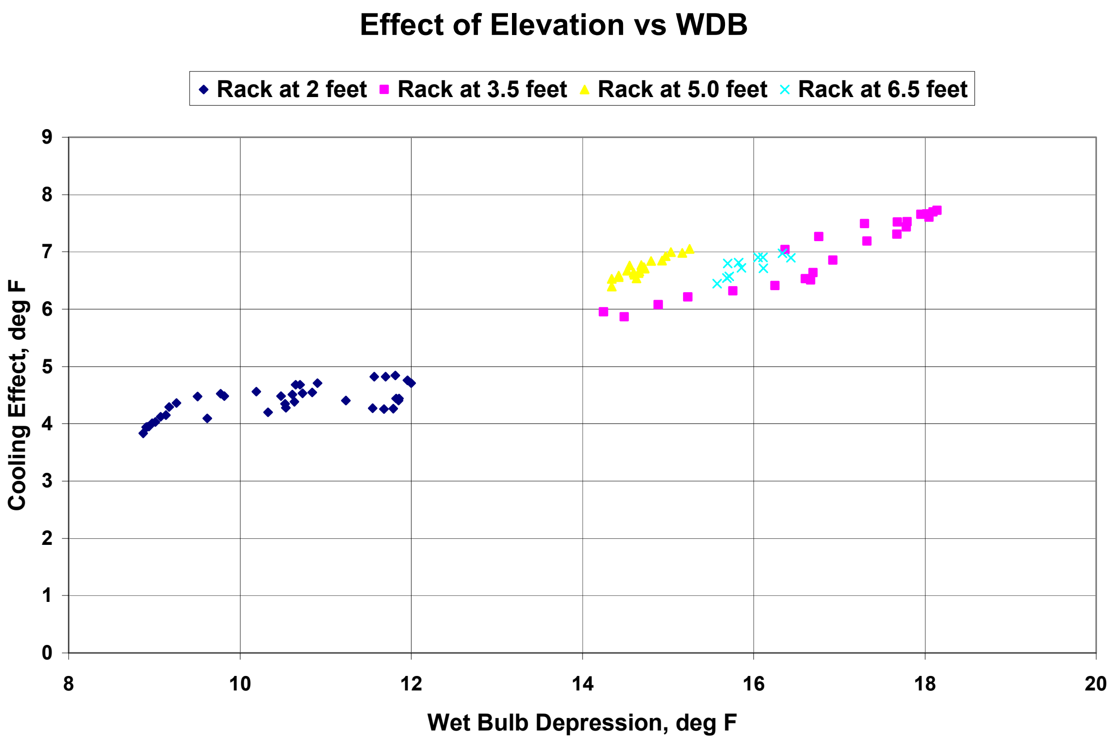


Figure 6-20
Effect of Elevation vs. WDB

Effect of Drop Size Distribution

Given the short residence time available for evaporation, it is beneficial to produce as fine a spray as possible. Other applications, such as gas turbine inlet cooling, achieve this with low-flow (0.05 to 0.1 gpm), high-pressure (2.0 to 4.0 kpsi) nozzles. For the larger flow rates required for ACC enhancement, such systems would require a large number of nozzles and be prohibitively expensive to install and maintain. Therefore, researchers tested the performance of different candidate nozzles to determine whether higher flow, lower pressure nozzles could provide acceptable drop size distributions and evaporation rates. They used three different nozzles, as tabulated with their characteristics in Table 5-1.

The changeover from one set of nozzles to another took several hours. Therefore, once installed, researchers left each nozzle set in place until they had completed all desired tests on those nozzles. Therefore, nozzle-to-nozzle comparisons are from runs widely separated in time and, in some cases, ambient weather conditions because the test period ran from late summer to mid-autumn. Furthermore, since the difference in the drop-size distributions among the nozzles is not great, the effect of slightly different spray rates or ambient conditions could easily obscure the effect of differing droplet size distributions.

Figures 6-21 through 6-24 present data from the three nozzles as cooling effect vs. wet-bulb depression. Each plot is for a narrow range of spray rates.

At the high (~20gpm) flow rates (Figure 6-21), there is no discernible difference among the three nozzles, with the possible exception of a slightly higher cooling effect with the "Fine—19 gpm" nozzle at the high (16.5 to 18° F) wet-bulb depression. However, the difference is no greater than the spread in any of the data sets and so is not likely to be significant.

At the intermediate (Figure 6-22; ~ 13 gpm) flow rate, the "Fine—13 gpm" data is in agreement with the data at the lower wet-bulb depressions as the performance at 13 gpm would be expected to level off at the higher wet-bulb depressions at a maximum cooling effect of about 8° F (see Figure 6-32 and related discussion).

The 10 gpm data (Figure 6-23) is again consistent with performance curves of the general shape Figure 6-32 displays. Some of the higher points of the "Swirl—11.5gpm" data set lie somewhat above the maximum cooling effect that would be calculated for that flow rate. This may be due to possible wetting of the fan exit dry-bulb temperature probes, which—if it occurred—would give an erroneously high cooling effect.

The low-flow rate data shows reasonable agreement between the "Coarse—8.4 gpm" and the "Fine—8.7 gpm" data sets. The "Swirl—6.3 gpm" appears to be low, but it is at or near the maximum cooling effect that could be achieved at that flow rate. These data lead to the conclusion that, for the range of nozzles and droplet size distributions tested, there was no observable systematic effect.

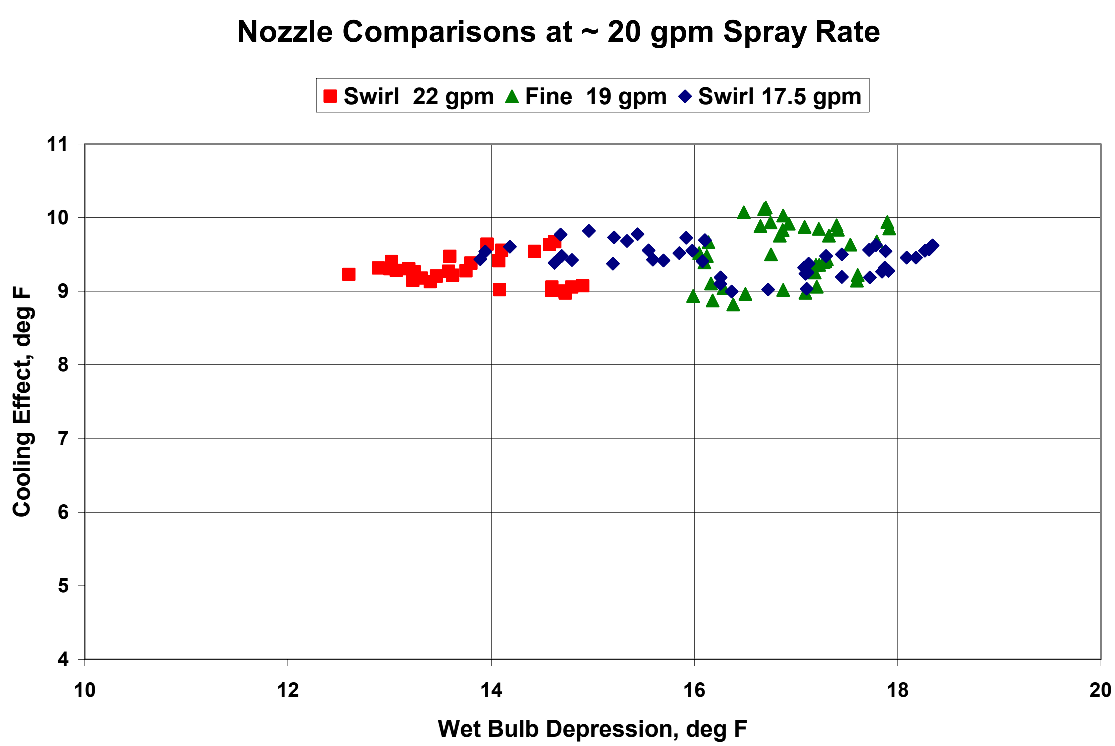


Figure 6-21
Nozzle Comparison at ~ 20 gpm Spray Rate

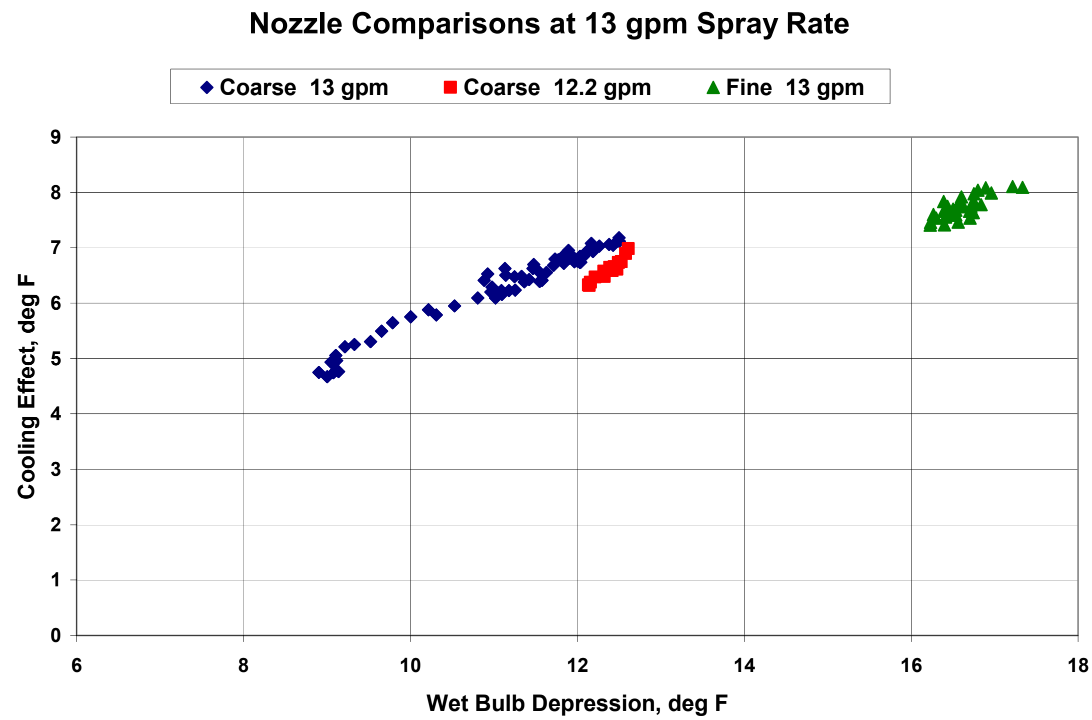


Figure 6-22
Nozzle Comparison at ~ 13 gpm Spray Rate

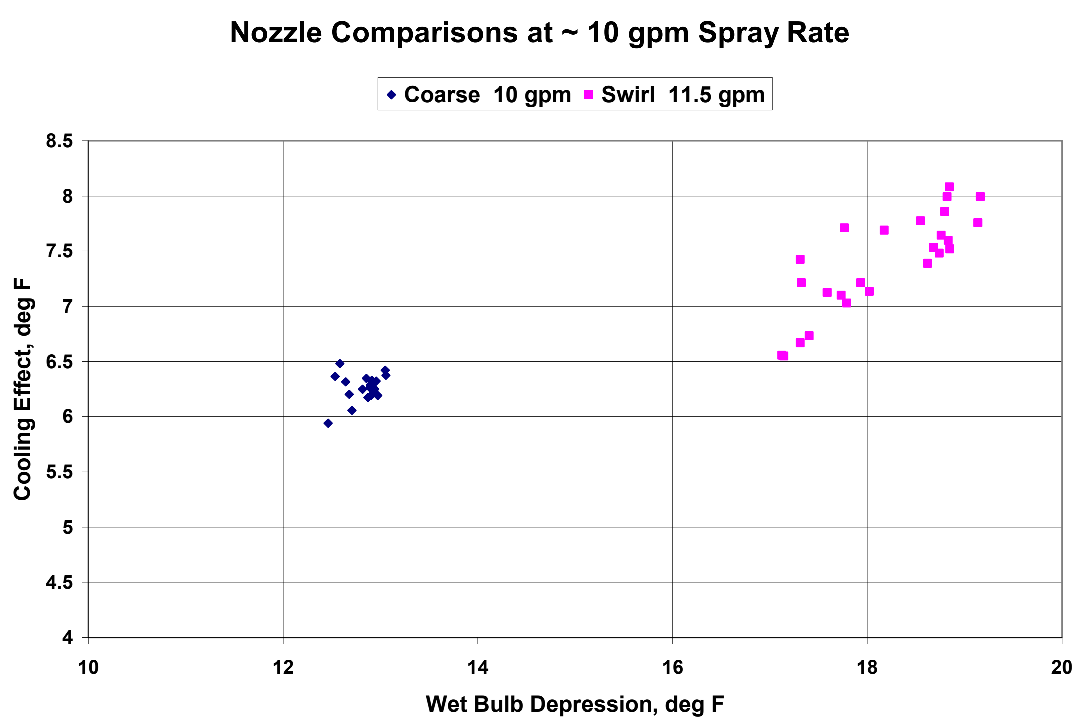


Figure 6-23
Nozzle Comparison at ~ 10 gpm Spray Rate

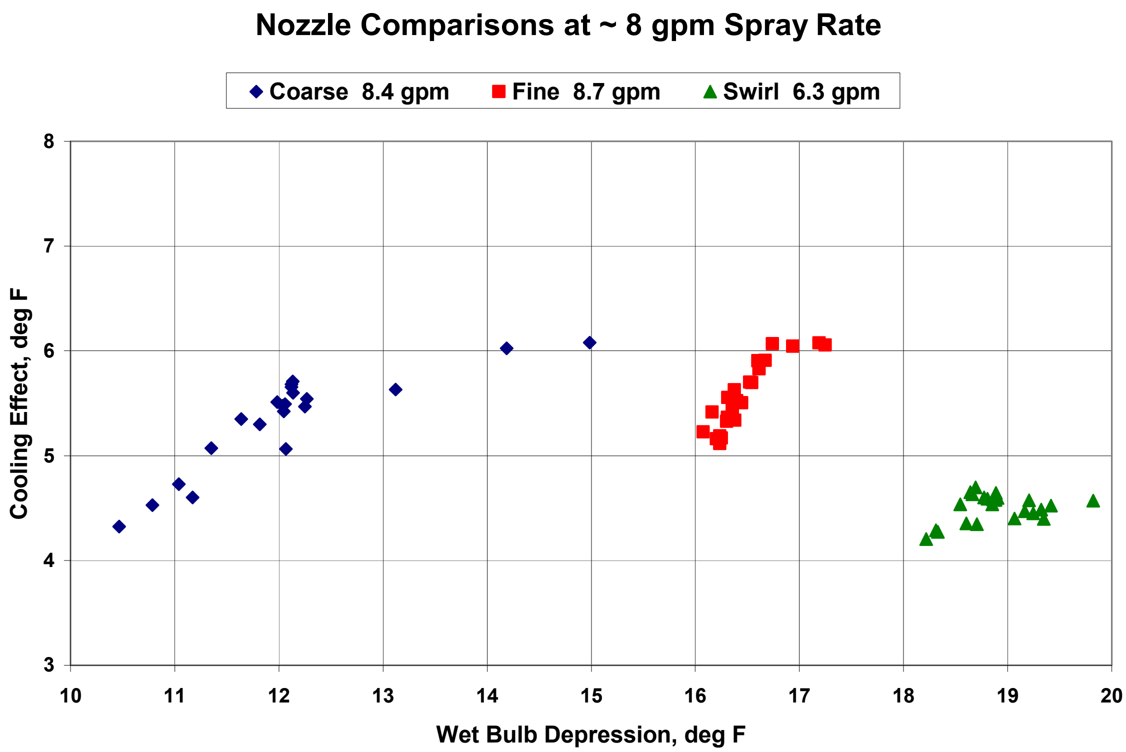


Figure 6-24
Nozzle Comparison at ~ 8 gpm Spray Rate

Operation With Demisters

As Section 3 discussed, one concern for the use of inlet spray cooling is the potential for harm to the finned tube surfaces from wetting by unevaporated droplets. One possible approach to address the problem is the use of demister panels between the spray nozzles and the finned tube bundles.

Researchers conducted a final set of tests at the end of the testing season in which they installed a panel of Brentwood CF-1900 counter-flow cooling tower fill on top of the screen just below the fan. Figure 6-25 shows a section of the fill. The inclined flutes through which the air passes turn the flow and cause the larger entrained droplet to hit the surface and be removed from the air stream. They then drain down the passages, counter flow to the air, and rainback to the ground beneath the fan. Figure 6-26 shows the demister panel in place on the test cell during operation of the sprays.

The results are shown in Figure 6-27 and 6-28 as cooling effect plotted against dry-bulb temperature (Figure 6-27) and wet bulb depression (Figure 6-28). Although the cooling effect is quite low (1 to 2° F) compared to previous results without the demisters, the test conditions were very unfavorable for evaporative cooling. The ambient temperature was in the range of 65 to 70° F and the wet-bulb depression was between 5 and 10° F.

At these ambient conditions, the cooling effect is reasonably consistent with extrapolated results at similar flow rates from data taken without the demisters but at higher temperatures and wet-bulb depressions. The use of demisters also appeared to increase the rainback as might be expected. However, with so little evaporation taking place, the rainback would have been high in any case compared to previous runs that were at the same spray rate but achieving much higher cooling effects. Since no demister operation data during hotter, drier weather are available, the results at this time are inconclusive and the use of demisters for the protection of the finned tubes surfaces merits further investigation.

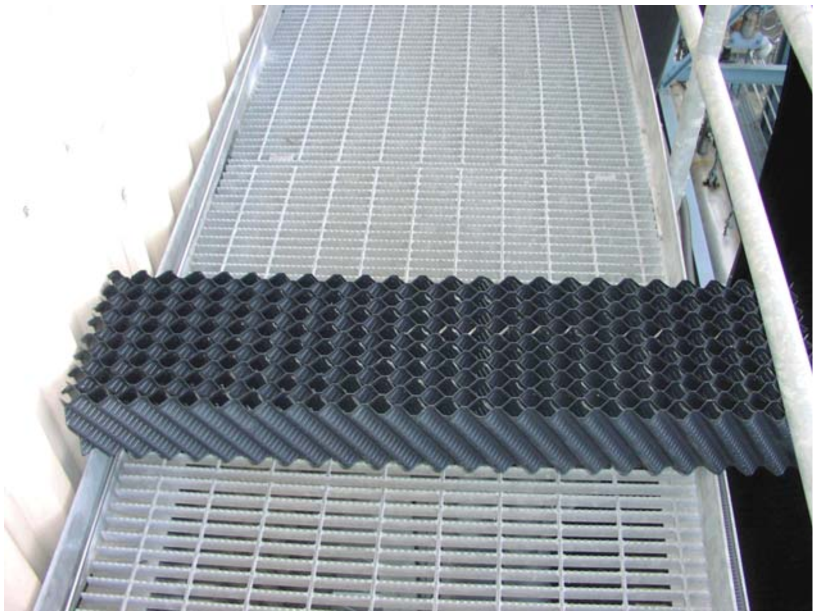


Figure 6-25
Section of Demister (Brentwood CF 1900 counter-flow fill)

Figure 6-26
Demister panels in place below fan

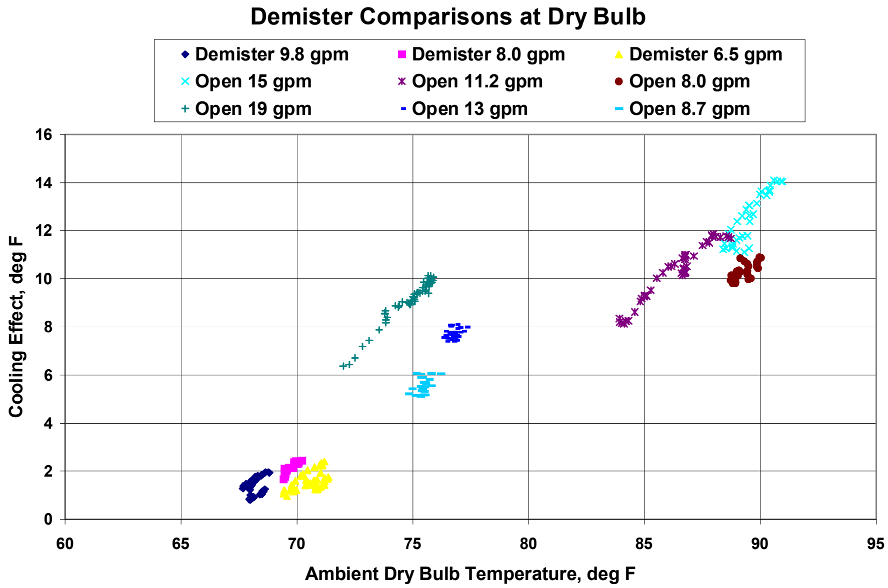


Figure 6-27
Demister Comparisons at Dry Bulb

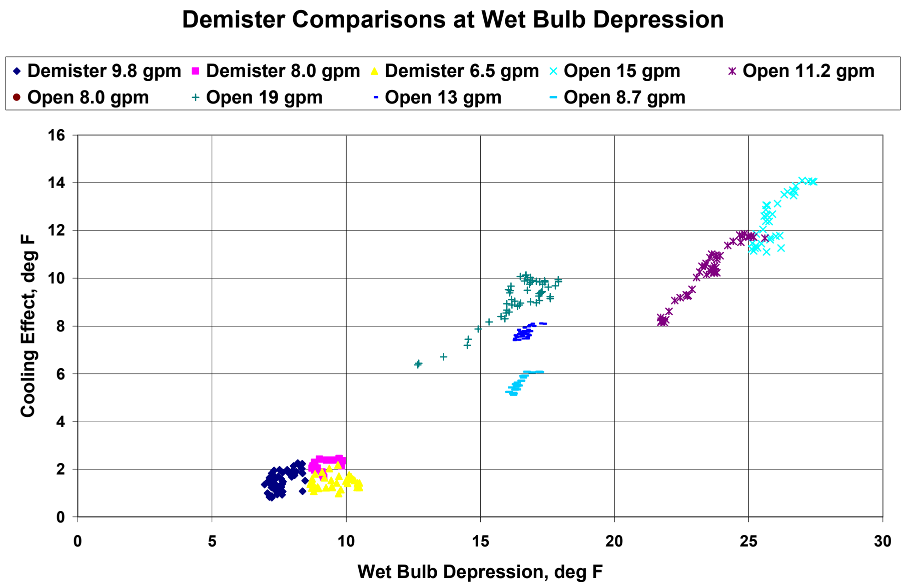


Figure 6-28
Demister Comparisons at Wet Bulb Depression

Performance Correlation and Design Rule

Based on the test results, the major variables that determine the cooling effect are the spray flow rate and the meteorological conditions. Secondary variables are droplet size distribution (determined by nozzle type) and residence time (determined by nozzle elevation in these tests). For generalized evaluation purposes, researchers developed a correlation of cooling effect with the major variables.

Physical Reasoning

The normal mass transfer mechanisms—the vapor pressure of the water at the droplet surface, the partial pressure of water vapor in the atmosphere, and the surface area of the droplets in the spray—drive the evaporation of droplets in an air stream. To a first approximation, the surface area is proportional to the spray flow rate (for similar droplet size distributions) and the driving force is closely related to the wet-bulb depression. Figure 6-29 displays an empirical, dimensional correlation between the cooling effect and the product of the wet-bulb depression (in degrees F) and the spray rate (in gpm).

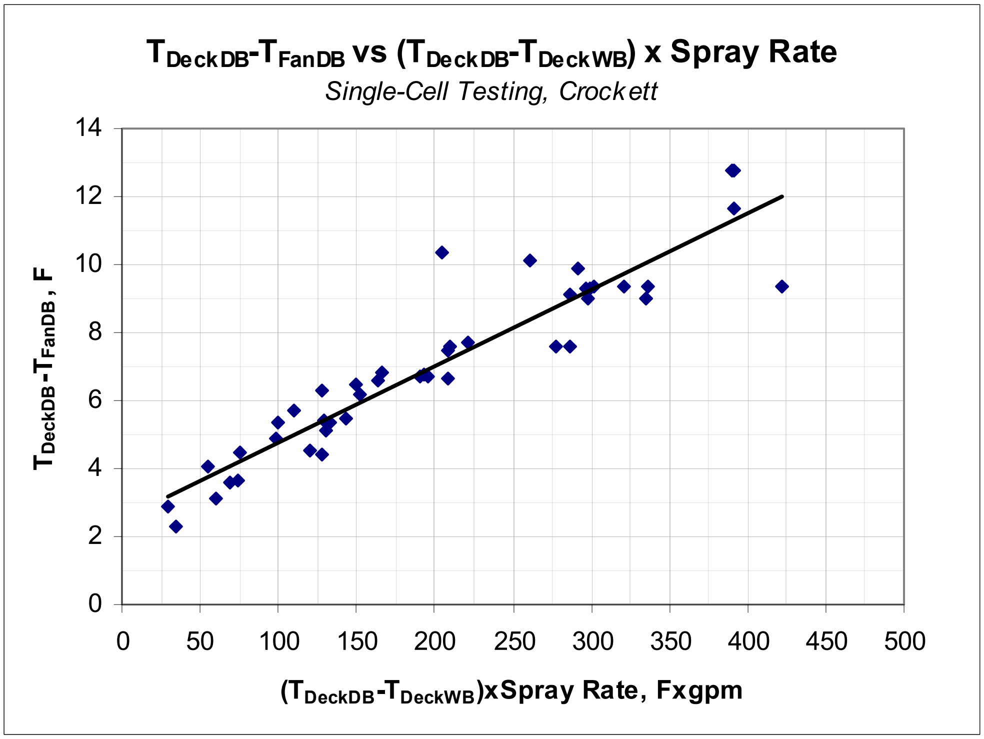


Figure 6-29
Correlation of Cooling Effect Data

The agreement is quite good with most of the data over a wide range of operating conditions falling within +/- 10 to 15% of a best-fit line.

Limits to Correlation

The form of the correlation shown in Figure 6-29 has fundamental limits, however. At a fixed spray rate, the cooling effect cannot exceed that amount of cooling that would be provided when 100% of the sprayed water evaporates. Similarly, at a fixed wet-bulb depression, the cooling effect cannot exceed the wet-bulb depression for any spray rate, no matter how large.

A simple analysis establishes maximum limits on the cooling effect. The airflow to ACE Cell B is 816,000 acfm. For this air flow, Figure 6-30 shows the spray flow needed to saturate the inlet air over a range of ambient dry-bulb temperatures for wet-bulb temperatures from 55 to 75° F. Figure 6-31 demonstrates that the spray flow required to saturate the air is determined solely by the wet-bulb depression for any dry bulb temperature within the normal range of ambient conditions.

Therefore, for all operating conditions, the maximum cooling effect can be determined simply as a function only of the wet-bulb depression and the spray flow rate. For any spray rate, the maximum cooling effect is equal to the wet-bulb depression up to that value at which the entire spray flow is evaporated, at which point the cooling effect can go no higher.

Figure 6-32 plots the maximum cooling effect for spray flows from 5 to 25 gpm and for wet-bulb depressions from 0 to 25° F, covering the range of test conditions for the field tests at Crockett.

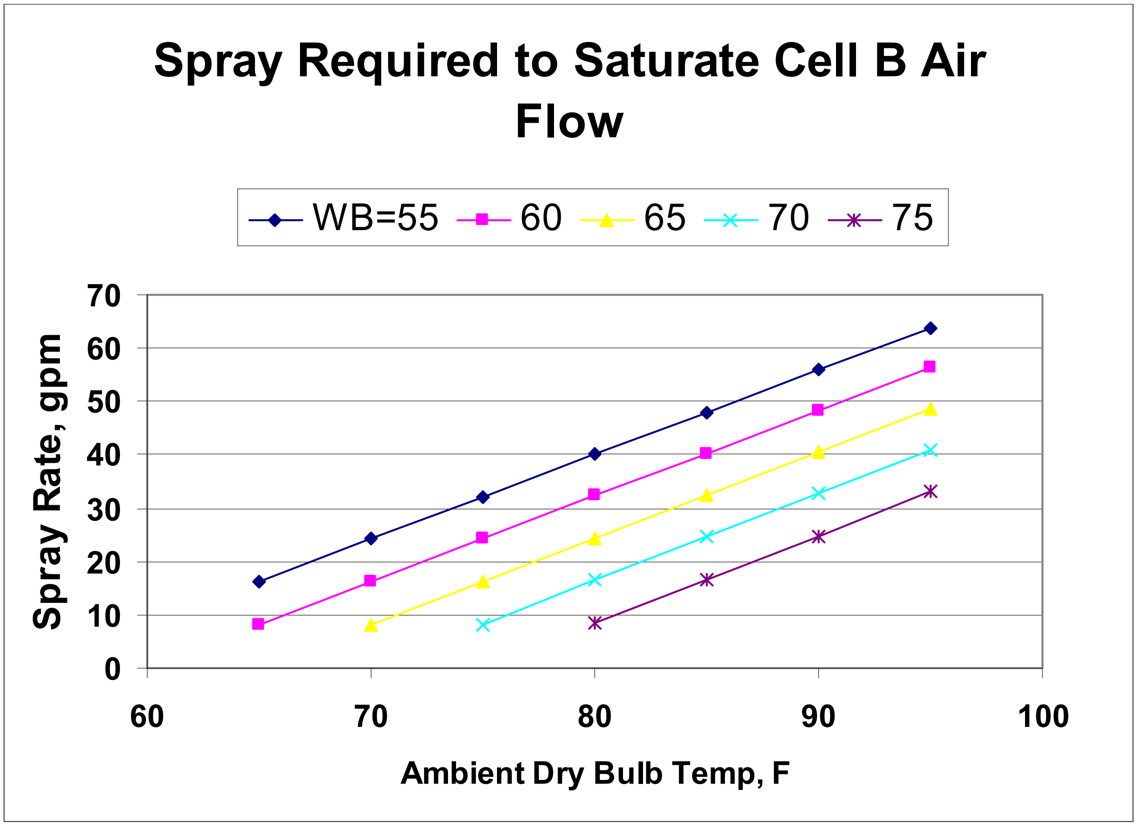


Figure 6-30
Spray Required to Saturate Cell B Air Flow

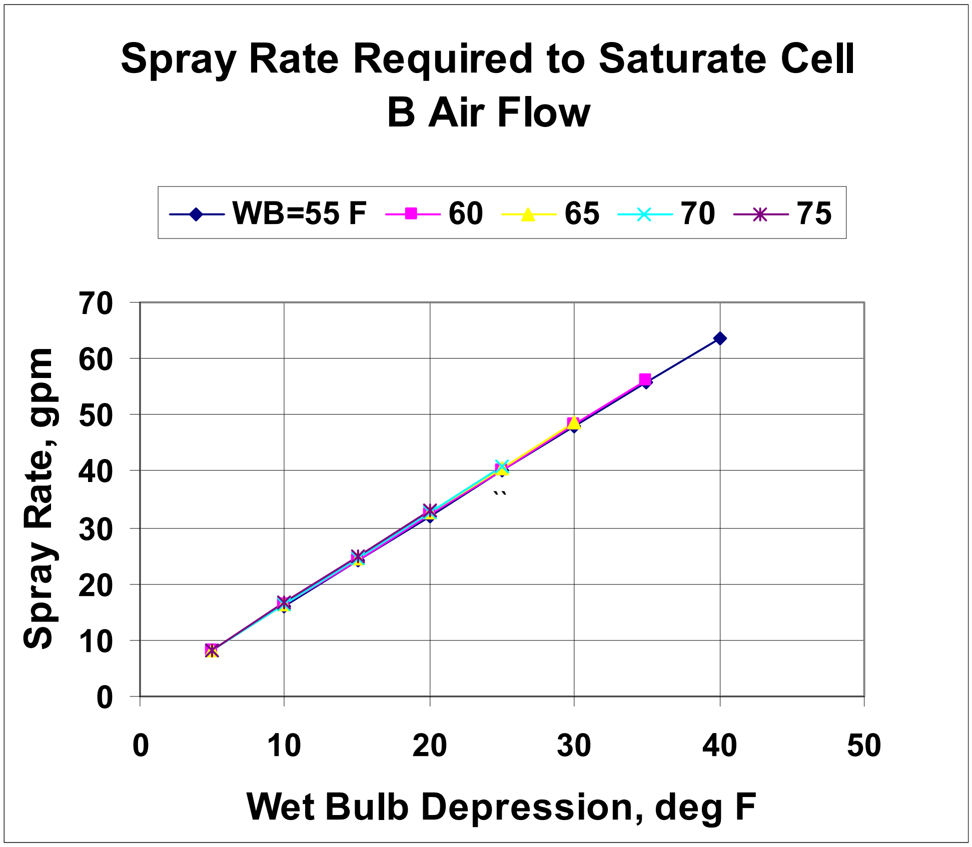


Figure 6-31
Spray Rate Required to Saturate Cell B Air Flow

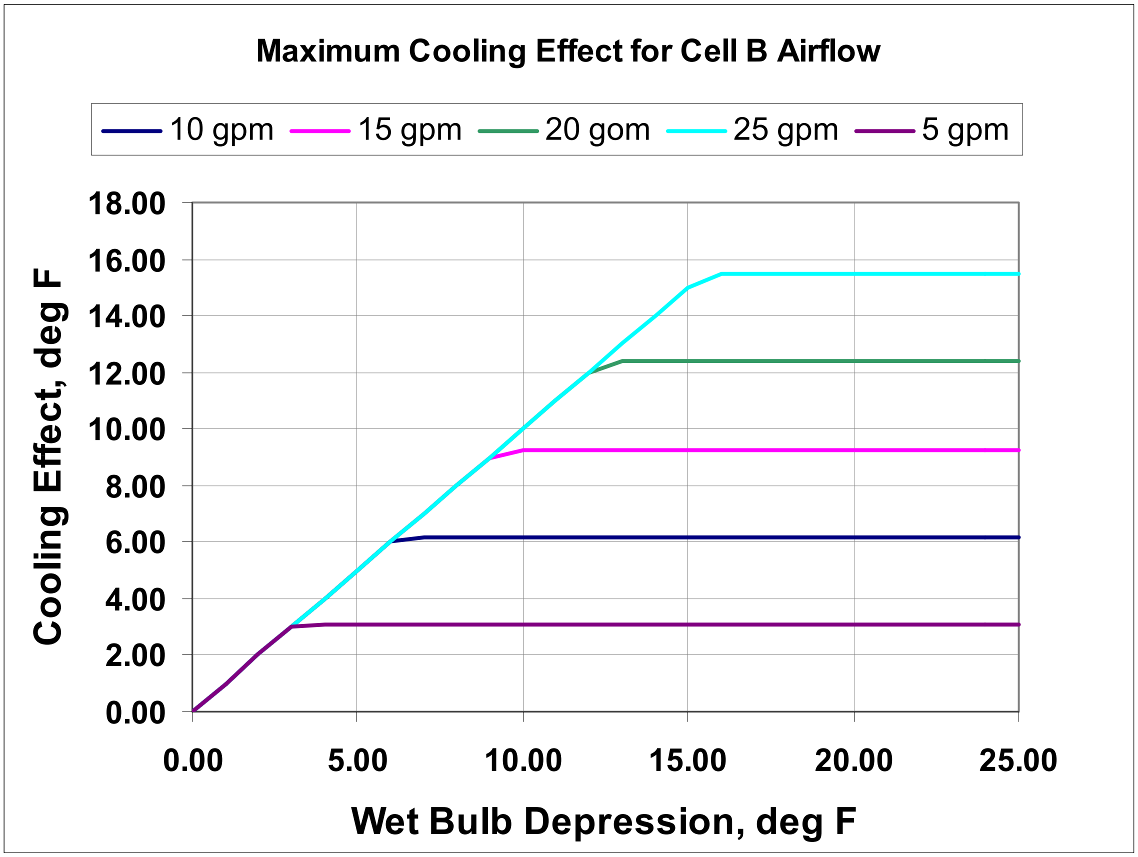


Figure 6-32
Maximum Cooling Effect for Cell B Airflow

Comparison With Data

The general form of the relationship, in which the cooling effect rises linearly with wet-bulb depression to a maximum and then flattens out, is seen in many of the individual test runs as, for example, in Figures 6-9, 6-12, 6-15, 6-16, and 6-21–6-24. Figures 6-33–6-35 show a direct comparison of test data against the limits developed in Figure 6-28 for spray flows of approximately 19, 13, and 8.7 gpm.

Examination of the comparisons suggests that the cooling effect of over 70% of the maximum is attained in essentially all cases. For ambient conditions at which spray enhancement would normally be considered, cooling effects of 85 to 90% of the maximum are readily attained.

Figure 6-33 plots the data at the high-flow rates along with the “maximum @ 20 gpm” line at 12.3° F. The maximum cooling effect for 19 gpm would be 11.7° F ([19/20] x 12.3 = 11.69). The data ranges from a cooling effect of 6.5 to 10° F corresponding to 56% to 85% of the maximum.

The data at both the intermediate flows (13 gpm; Figure 6-34) and low (8.7 gpm; Figure 6-35) show cooling effects at essentially the maximum levels to be expected for these flow rates.

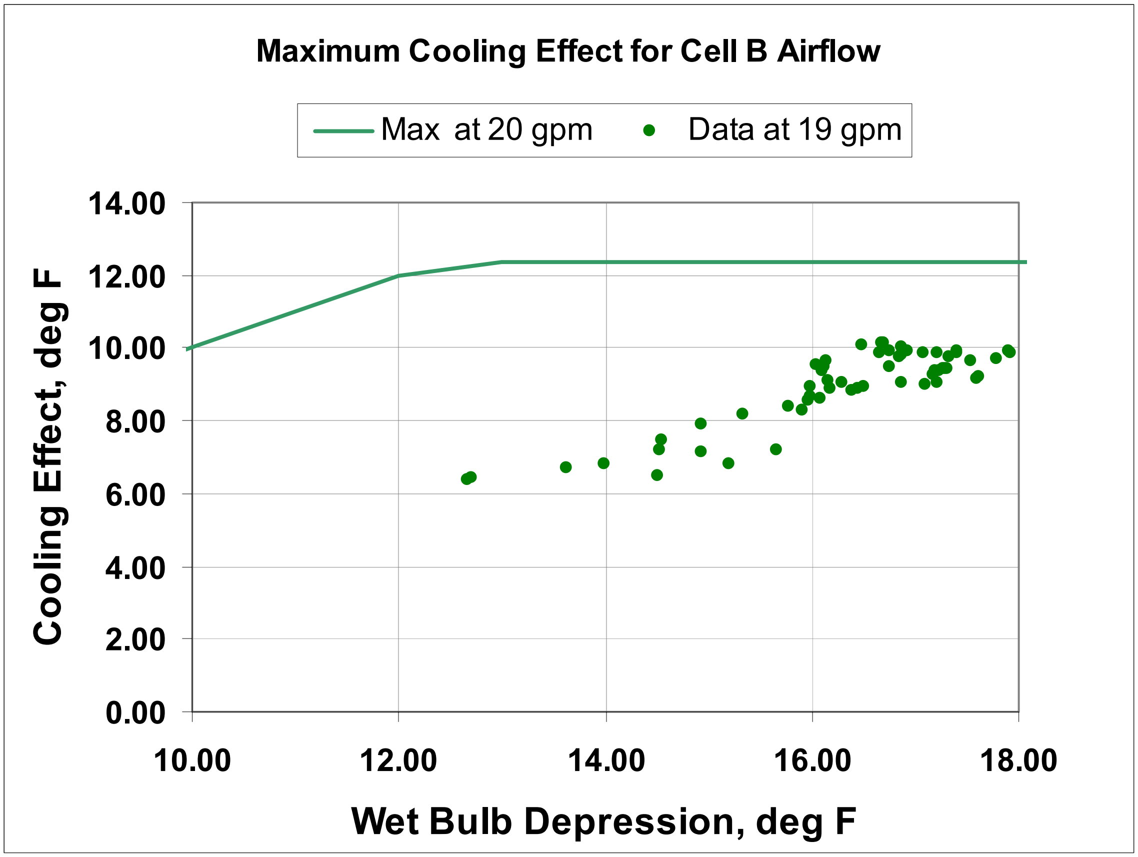


Figure 6-33
Cooling Effect Data Compared to Maximum Cooling Effect (19 gpm)

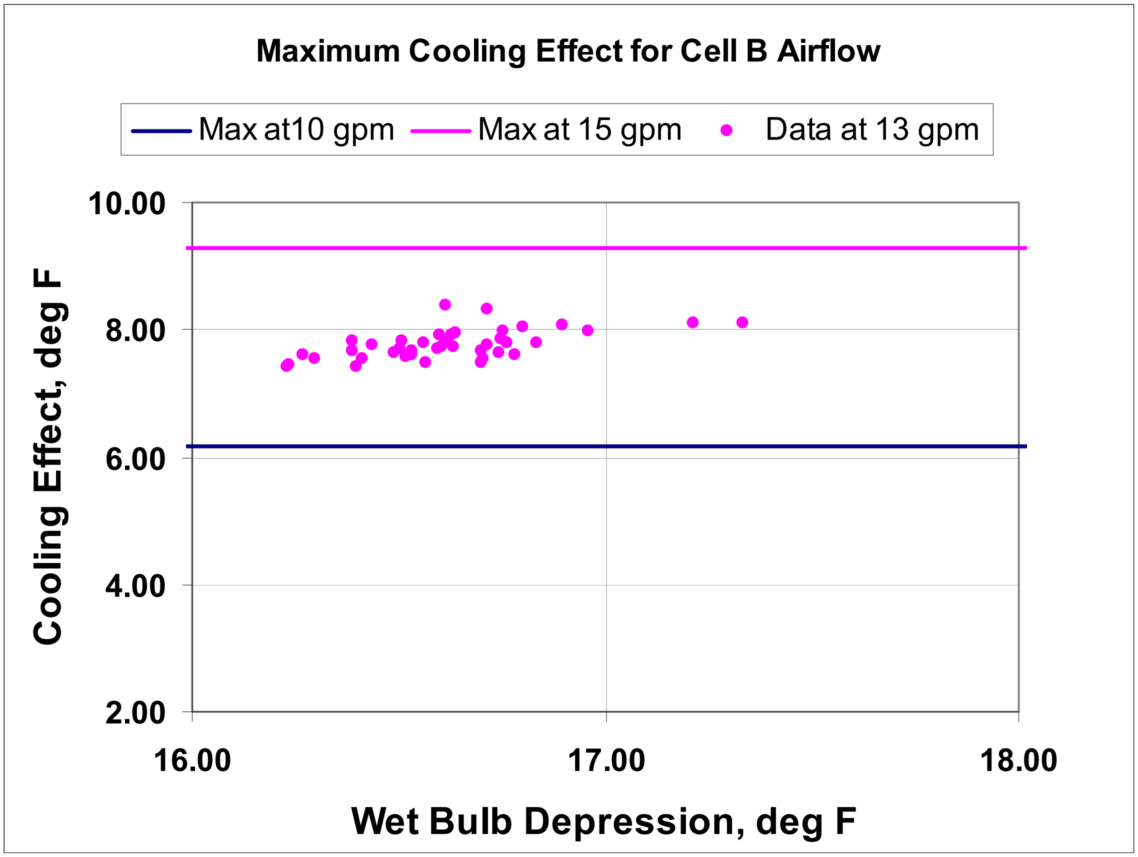


Figure 6-34
Cooling Effect Data Compared to Maximum Cooling Effect (13 gpm)

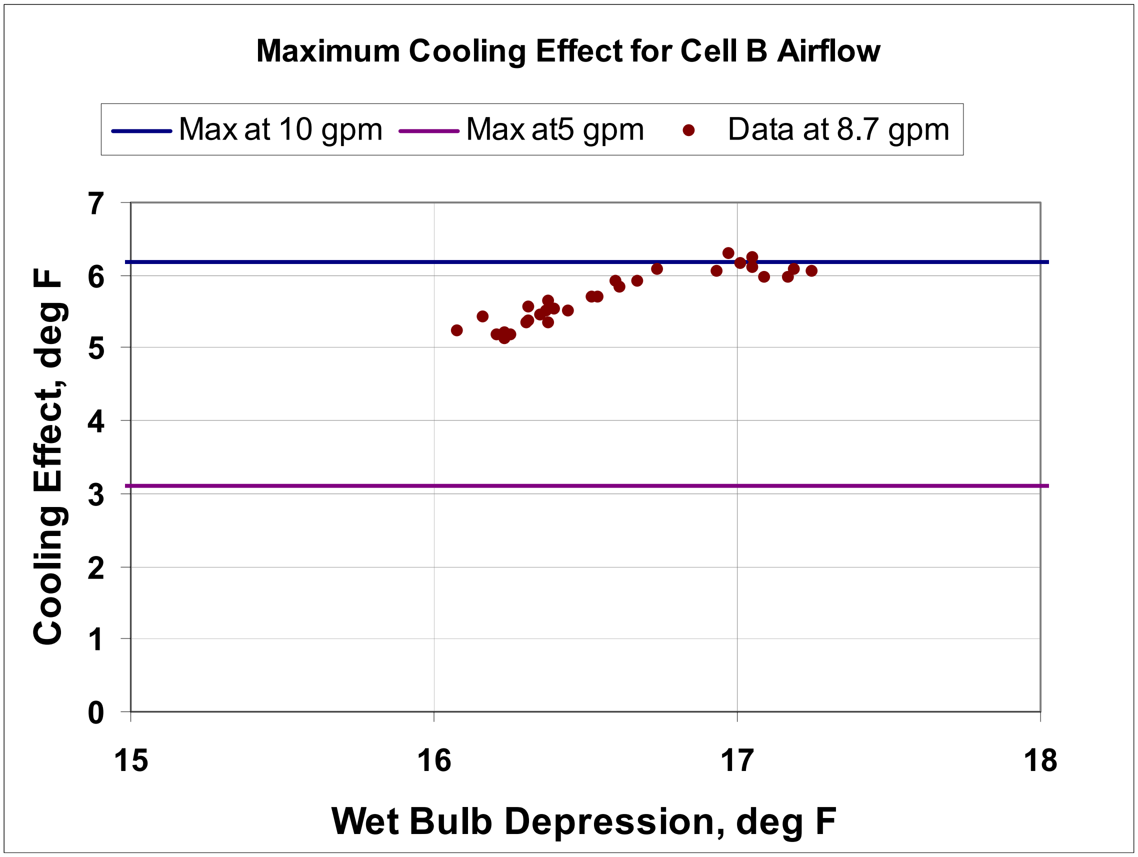


Figure 6-35
Cooling Effect Data Compared to Maximum Cooling Effect (8.7 gpm)

7
COST BENEFIT ANALYSIS

The following section provides information on the costs and benefits of a spray enhancement system for air-cooled condensers. It reviews the various cost elements of a system and provides estimates for a range of system sizes. The benefits section illustrates the methodology for estimating the amount of additional output and efficiency improvement to be expected from the use of spray enhancement.

Costs

Researchers developed a methodology for the design and estimation of a full-scale inlet air-cooling system for an air-cooled condenser. While the design of an inlet air-cooling system is site specific, there are a number of common factors, considerations, and components for each system, as follows:

- Specification of design inlet air temperature and desired air cooling
- Expected spray water cost, quality, and quantity
- Required protection, if any, of the steam condensing coils from the spray cooling water (for example, salt deposition or corrosion)
- Impacts, if any, of spray water carryover or fall out from the spray arrays in the immediate vicinity of the ACC
- Prevailing wind conditions and potential screening requirements (to reduce wind shear effects on spray nozzles)
- Prospective locations for a pump skid, possible spray water storage and pipe routing options
- Spray system control

An example worksheet in Appendix C illustrates the method and items included.

Design and Estimating Tools—General Cooling System Specifications

The worksheet (Appendix C) serves as a starting point for estimating the piping requirements, general design, and approximate cost of an inlet-air, spray-cooling system for an air-cooled condenser. It is an example of the type of analysis possible to evaluate spray system costs. Developing an estimate requires the following inputs:

- Design inlet air dry-bulb and wet-bulb temperatures

- Desired reduction in dry-bulb temperature (limited by the prevailing wet-bulb temperature)
- Number of cells/fans and the air flow rate for each
- Overall air-cooled condenser dimensions

Design and Estimating Tools—System Design Assumptions

Cost analysis took into account the following parameters:

The **design cooling effect or reduction in inlet dry-bulb temperature** is taken as 70 percent of the wet-bulb depression. This is consistent with the test results Chapter 6 discussed and is conservative for the hot, dry conditions at which plants would use spray enhancement.

The **amount of spray water** required is calculated from a heat balance (that is, the heat removal from the air stream is equal to that taken up by the evaporating spray droplets). The example in Appendix C provides a “safety factor” of ten percent—that is, the design spray rate is 10% greater than the minimum required for the design cooling effect.

The **number of nozzles** is based on an assumed flow of 0.5 gpm per nozzle.

The **pump horsepower** requirements are calculated using a 350 psig delivery pressure rounded up to the next commercially available motor/pump size. It is recommended that the pump be backed up with a standby.

Piping requirements are estimated using the following assumptions:

- Total length of the piping is a function of the number of cells and the air-cooled condenser dimensions.
- Piping is assumed to be stainless steel and designed for 350 psig service. Installed costs include labor and materials. Installation includes pipe hangers, assumed to be easily attached to the existing ACC structure. No additional structural support or requirements are assumed. Design velocities of 6–10 ft/second are used
- The spray nozzle array could take a number of different forms, such as nozzles oriented in a circular pattern or the square matrix of nozzles that was deployed at Crockett. Clearly, the cost estimates for each may vary significantly and will be site specific in nature.

The **Control system** for an ACC inlet-air cooling system could range from manual operation to a sophisticated computer-based optimization and control system. One of the more cost effective and versatile control approaches for a system of this type is a PLC (Programmable Logic Controller). A PLC can carry out several applications such as monitoring ambient dry- and wet-bulb temperatures and wind conditions and controlling spray water flow rate. Users could tie it into central plant monitoring and control instrumentation. Basic PLC costs run approximately $10,000 to $15,000 depending upon the PLC’s sophistication and features. Programming costs are typically $5,000 for a basic system.

If a user desires **ambient psychrometric measurements** as input to a PLC**,** it will add $6,000 to $10,000 for a dedicated meteorological station.

Optimization

The design considerations for such a system could also include optimization of the spray water rates and plant heat rate improvements. Further, it could also incorporate modulation of spray rates based on water costs and heat rate improvements. A PLC interfaced to the main power plant control system could easily accommodate and drive these considerations.

That the exact number and location of spray nozzles for a particular application is still a part of the "art" that is evolving from research and testing programs of this type. The ideal design would be characterized by the following:

- As few nozzles as possible at a distance that is minimally 10 ft (3m) from the fans, but far enough away to effect complete evaporation of the sprays prior to the fans or steam condensing coils
- Maximum evaporation and minimal water loss (a 10 percent safety factor should be incorporated in the calculation of required spray water)
- Pipe routing that results in minimal lengths and diameters, as much as possible

Table 7-1 gives some approximate costs using the approach described.

Table 7-1
Example Inlet Air Spray Cooling System Cost Estimates

Cooling System Size	Reduction in DBT (F)	Spray Water Flow (gpm)	Estimated Cost ($)
15 Cells	20	665	$280,000
5 Cells	25	277	$240,000
30 Cells	30	1994	$600,000

Benefits Analysis

It is easier to understand the benefits of ACC inlet-air cooling in the context of the design point optimization procedure for air-cooled systems. The selection of the design point represents a trade-off between initial capital costs of the ACC and the costs imposed on the plant through reduced efficiency and capacity loss that use of dry cooling may impose.

The basic design decision is the selection of the ambient temperature at which a desired turbine backpressure can be maintained. One common choice is to require that a turbine backpressure of 2.5 in Hga be maintained at the annual average temperature at the site. The consequences of this may be significant performance limitations during the hottest months of the year, depending on site meteorology and turbine characteristics. The following example will illustrate the trade-offs.

Table 7-2 gives meteorological data for a hot, arid site in California. The annual average dry-bulb temperature at the site is 72° F. To achieve a turbine backpressure of 2.5 in Hga requires that the steam condensing temperature be maintained at 109° F. The difference (condensing temperature minus ambient dry-bulb temperature) is known as the Initial Temperature Difference or ITD and is the design variable that determines the size and price of air-cooled condensers.

Table 7-2
Site Specifications

Characteristic	Hot Arid Site
Location	**Southern California**
Elevation (ft)	390
Temperature, Dry Bulb	
T_{avg}	72
T_{max}	117
$T_{1\%}$	111
$T_{2\%}$	109
$T_{5\%}$	106
Temperature, Wet Bulb	
$T_{wb\ avg}$	63
$T_{wb\ max}$	79
$T_{wb\ 1\%}$	78
$T_{wb\ 2\%}$	77
$T_{wb\ 5\%}$	76

Figure 7-1 displays typical capital costs for air-cooled condensers capable of condensing 1.0 million pounds of steam per hour (the approximate steam flow for the steam portion of a 500 MW combined cycle plant) for a range of ITD’s. For the example described above, the ITD is 37° F (109° F – 72° F).

Figure 7-1 shows a performance curve for a typical air-cooled condenser sized to a design ITD of 37° F. At temperatures above the design value of 72° F, the backpressure that can be maintained rises and reaches 6.5 in Hga at 110° F. At the maximum temperature of 117° F at the example site (see Table 7-2), the backpressure would approach 7 in Hga, which is close to the point at which many plants must reduce load in order to protect the blades at the backend of the low-pressure turbine.

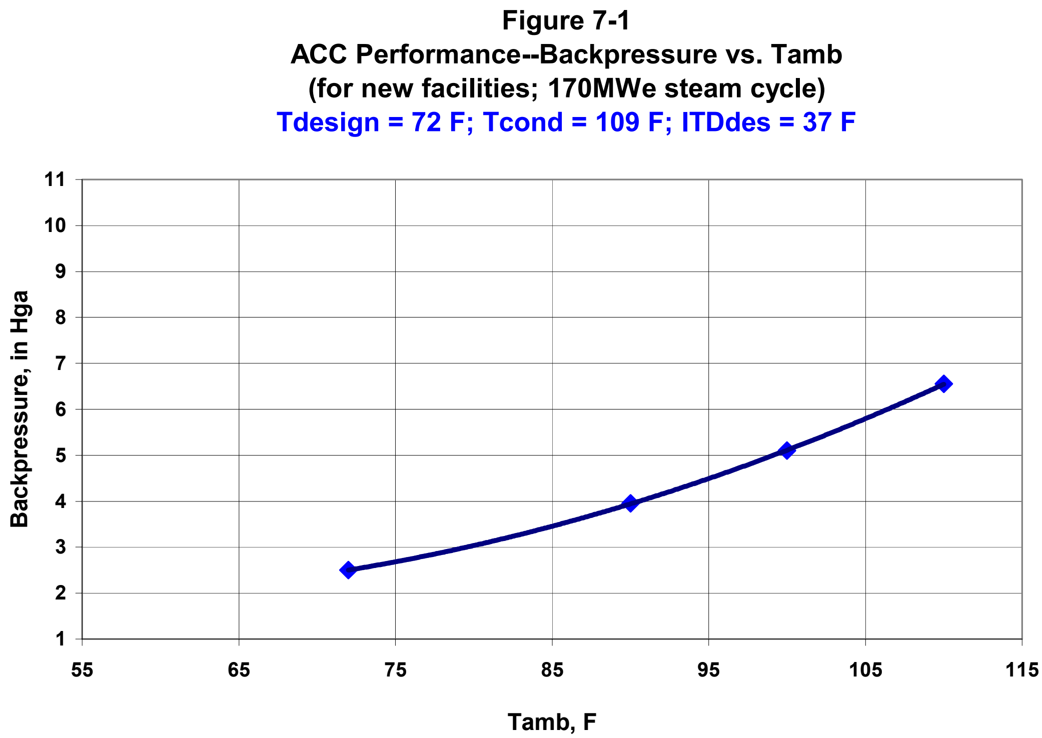


Figure 7-1
ACC Performance – Backpressure vs. Tamb

Figure 7-2 is a temperature duration curve indicating the number of hours per year any given temperature level is exceeded. For the site illustrated, the temperature exceeds 100° F for nearly 1000 hours per year.

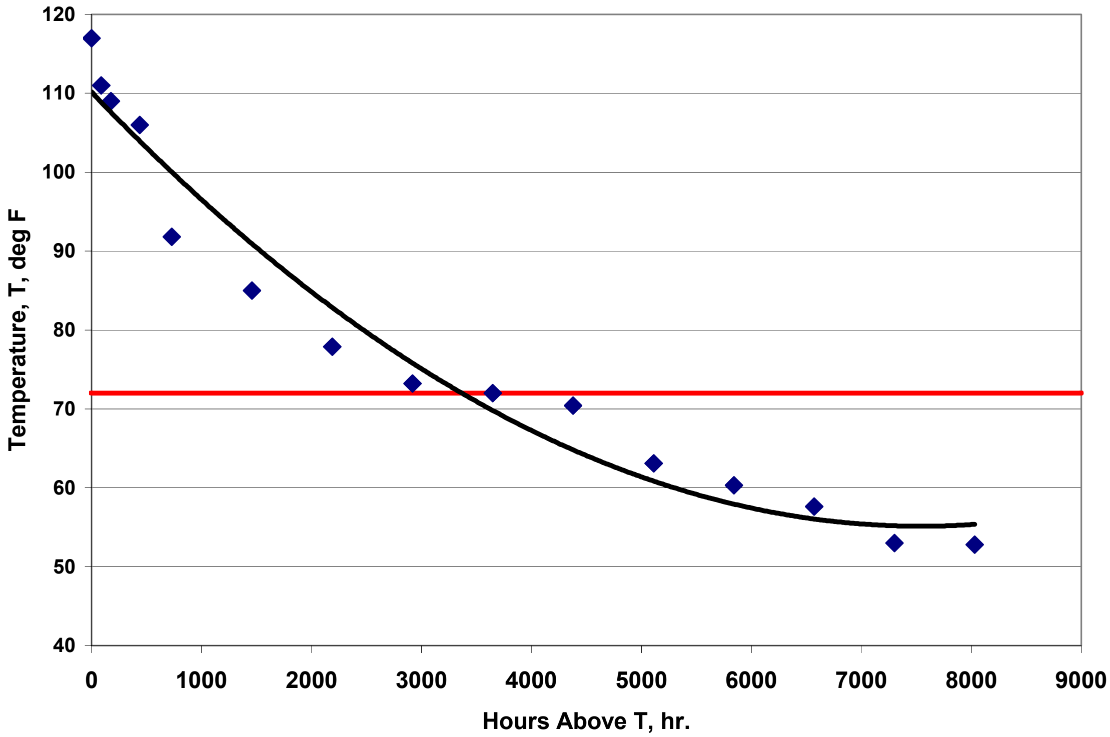


Figure 7-2
Temperature Duration Curve

Finally, Figure 7-3 displays a typical turbine heat rate curve showing the reduction in plant output as the turbine backpressure rises above the design point.

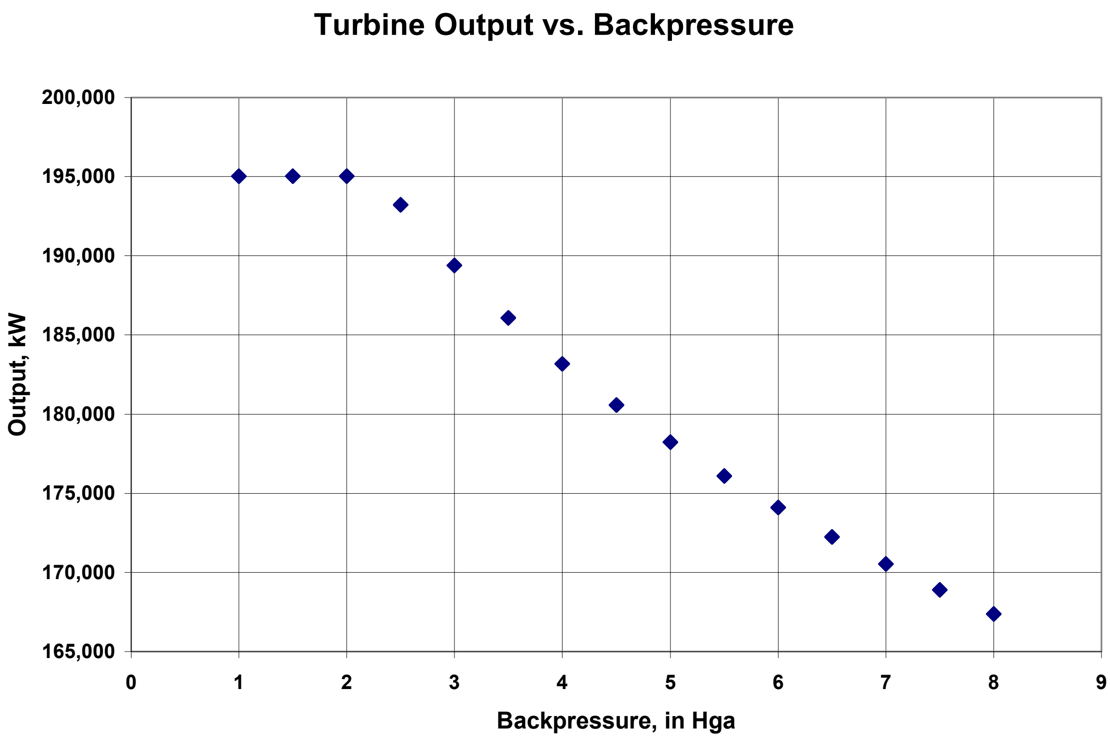


Figure 7-3
Turbine Output vs. Backpressure

For this example, an increase in backpressure to five in Hga incurs a lost output of approximately 17 MW.

From Figure 7-1, a 5 in Hga backpressure will occur when the ambient temperature reaches 100° F and from Figure 7-2, this will occur for approximately 1000 hours per year. These conditions result in a minimum output loss of 17,000 MWh per year, valued at least $500,000 at an average power price of $30/MWh. Given that the losses occur at the hottest hours of the year when demand is the greatest and prices have historically spiked to much higher levels, the lost revenue could be many times that amount.

It is possible to estimate the amount of this lost output that a plant might recover with a spray enhancement system. From Table 7-2, the wet-bulb depression (the maximum inlet air cooling theoretically achievable) can be obtained for a range of ambient dry-bulb temperatures. Figure 7-4 displays a plot of wet-bulb depression vs. ambient dry-bulb temperature.

Figure 7-4
Wet Bulb Depression vs. Dry Bulb Temperature

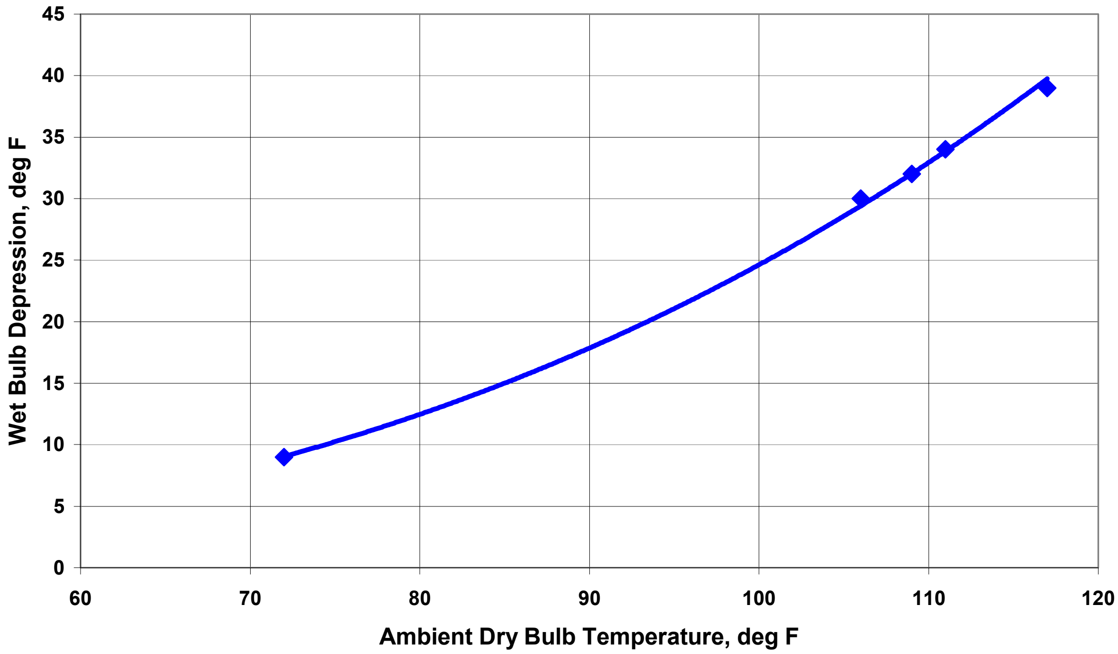


Figure 7-4
Wet Bulb Depression vs. Dry Bulb Temperature

From the field test results at Crockett that Section 6 analyzed, it is reasonable to assume that during these hot, dry conditions, the spray enhancement system could achieve a cooling effect of at least 75% of the wet-bulb depression or over 18° F for all of the 1000 hours above 100° F.

Referring again to Figure 7-1, this would reduce the backpressure from 5 to 3.5 in Hga (at the 100° F ambient condition). Figure 7-3 shows that this would restore the turbine output to 186 MW, recovering approximately half of the lost output. The recovered energy, if again valued most conservatively at $30/MWh, is worth over $250,000 in a single year. If valued at prices closer to the historical levels for peak demand periods, savings could easily exceed $1 million in one year.

This benefit compares favorably with the costs for the spray enhancement system, which Table 7-1 estimates and displays. The ACC size used in this example corresponds to the 30-cell case in Table 7-1 with an estimated cost of $600,000. Therefore, a plant could recover the cost of the spray enhancement system well within the first year of its operation and would be able to provide power when it is most needed.

Water Requirements

Water use rates depend on the site meteorology and the number of hours per year that spray enhancement is used. The airflow to an ACC is approximately 250,000 cfm per MWe. Assuming a plant uses the spray enhancement whenever the ambient temperature goes above 90° F and sprays sufficient water to hold the inlet temperature at 90° F, the instantaneous spray rate is approximately 6.5 to 7.5 gpm per MWe. This corresponds to about 60% of the instantaneous evaporation rate for a wet cooling tower but for only 100 hours per year. Therefore, on an annual basis, the water use is less than 7% that for wet cooling.

8
SUMMARY AND RECOMMENDATIONS

Summary

- As part of a previous study (Comparison of Alternate Cooling Technologies for California Power Plants, EPRI and CEC, #83362), researchers determined that heat rate and capacity penalties during the hottest hours of the year were the major drawback to the use of dry cooling at many locations in California.
- Researchers showed that the allocation of cooling water in modest quantities (less than 10 to 15% of what would be required annually for the use of recirculating wet cooling) reduced the hot day penalties by 50% or more.
- Of the several methods that have been proposed for "hot day" enhancement of dry cooling, researchers selected the use of sprays to cool the inlet air for development and testing based on its simplicity, low capital cost, and projected return on investment.
- Spray system design criteria included low-pressure (< 300 psia) operation, moderate to high flow rates per nozzle (>0.1 to 0.4 gpm/nozzle), fine sprays (< 50 μ Sauter mean diameter), and low construction material cost.
- Researchers evaluated fifteen nozzles and three demister designs in laboratory tests.
- Researchers selected three nozzles and one demister for further testing at a full-scale cell of an air-cooled heat exchanger at an operating power plant (Crockett Cogeneration in Crockett, California).
- Field test results yielded the following general conclusions:
  - The cooling effect ($T_{amb} - T_{inlet}$) was a strong function of ambient wet-bulb depression ($T_{amb\ dry\text{-}bulb} - T_{amb\ wet\text{-}bulb}$) and spray flow rate.
  - The effect of spray droplet size distribution and nozzle location (droplet residence time) was discernible but typically minor.
  - The attendant cooling effect ranged from 60% to nearly 100% of the prevailing wet-bulb depression, depending on spray rate and ambient conditions.
  - At conditions where the use of spray enhancement would most likely be considered (T_{amb} > 90° F and relative humidity < 40%), a cooling effect of 80% or greater of the wet-bulb depression could be expected.
- A simple cost-benefit analysis consisting of a generalized system design procedure using costs based on industry "rules-of-thumb" and a simplified estimate of the expected increase in plant efficiency and output at a hot, arid site suggested a pay-back period of one year or less.

Remaining Issues and Recommendations for Future Work

The primary issue facing the successful use of spray enhancement of dry cooling system performance is that of minimizing or eliminating unevaporated spray. Water droplets that do not evaporate in the air stream have the following effects:

1. Contribute nothing to the cooling effect
2. Pose potential harm to the ACC finned tubes through scaling, fouling, or corrosion
3. Create unwanted rainback under the unit
4. Impose an additional water demand at the plant for no benefit contrary to the original result of choosing a dry cooling system

Further work to address this question and to assess the effectiveness and cost of the several proposed approaches is recommended in four areas.

Nozzle Array Optimization

Researchers chose the placement of the nozzles on an elevated rack under the test cell for these tests to enable quantitative measurement of the spray system performance. It was necessary to entrain as much of the sprayed water as possible into the instrumented cell in order to obtain a meaningful heat balance. With the wind conditions at the test site, the elevated rack configuration was the only possible choice.

However, for an operating system on a full-scale ACC, other considerations pertain. As long as some reasonable distribution of the spray across the several cells of an ACC can be achieved, it does not matter which cell the spray from any particular nozzle enters. ACC's generate a significant amount of their own local wind pattern and nearly all of the air entering the region under the cells is entrained by some part of the unit. Therefore, the introduction of a sufficiently fine spray into the region virtually guarantees its entrainment, and location of the nozzles at about two-thirds of the way up to the fan inlets and around the periphery of the unit may provide the best combination of rapid entrainment and extended residence time. Such an arrangement may also be preferred for convenience of installation, rack, and nozzle maintenance and for fan removal and maintenance when required.

A field test at a full-scale operating plant in conjunction with a modeling effort using advanced computational fluid dynamics code could result in valuable design guidelines.

Advanced Nozzles or Other Spray Devices

It is inevitable that there will be some degree of unevaporated spray, which will eventually fall as rainback onto the area beneath the fans. This results from several sources, primarily from the impingement of spray droplets on structural members, spray racks and piping, nozzle pintles and the fan blades and shroud. Drainage from all these elements occurs in large (> 1000 μ) droplets that fall to the ground with little further evaporation. In most instances, this is not a large amount compared to the spray rate. During operation at spray rates of 20 gpm for 3 to 4 hours (a total spray amount of 4,000 to 5,000 gallons), the apparent accumulation on the ground was no more than a few hundred gallons. Nonetheless, for full-scale operation where the spray rates might be in the range of 1000 gpm lasting for six hours or more, the amount of rainback accumulating beneath the unit could be quite large. If this constitutes either an environmental or a "housekeeping" problem requiring collection and recycle, it would add to both the capital and O&M costs of the system. Advanced spray devices may address this problem.

While reasonable cooling effects were obtained with the nozzles chosen for test, improved spray devices would enhance the thermal performance of the system and reduce both the rainback and the potential for wetting the finned surfaces. Two avenues may be explored:

i. Industrial nozzle vendors claim recent advances in both colliding jet and internal swirl designs that may produce finer mist at the pressure and flow levels desired for the system. Tests of the latest developments should be pursued.

ii. EPRI is investigating a number of proposed advanced devices including a crystal nebulizer of the type developed for medical applications, a rotating screen device of the type used for aerial spray applications (such as crop dusting), and a nozzle-in-fan concept.

Water Purification and Management

Even at the most favorable test conditions some liquid droplets reached the inlet plane of the finned tube bundles. Only high-purity condensate was used as spray water for these tests. For designs in which similarly high-purity water is used, a separate water treatment system will need to be engineered and provided, since the spray rate for a full-scale application would more than likely exceed the excess capacity available in the plant make-up water system. In addition, the rainback problem discussed in Item #2 above may well lead to a recycle/reclamation requirement. While the technology is straightforward, the system will generate a brine stream and likely a sludge for which discharge and disposal provisions must be made. The cost of this water management system will add substantially to the cost of the base system and the trade-offs should be evaluated to support future economic design efforts.

Droplet Capture and Return

An alternative to the use of high-purity water for the spray is the use of demisters to ensure that no liquid droplets reach the finned tube surfaces, thus permitting the use of "raw" water. However, this approach might reduce the cooling effect and the water utilization efficiency by taking small droplets out of the flow early, preventing further evaporation and cooling. They may also increase the rainback problem if the collected droplets are drained into the area below the collection plane.

The use of design configurations similar to cooling tower fill in cross-flow towers may allow the positioning of demisters immediately upstream of the finned tube bundles and the use of a collection tray at the bottom edge of the demister for recycling of the unevaporated liquid.

A
LABORATORY TEST RESULTS

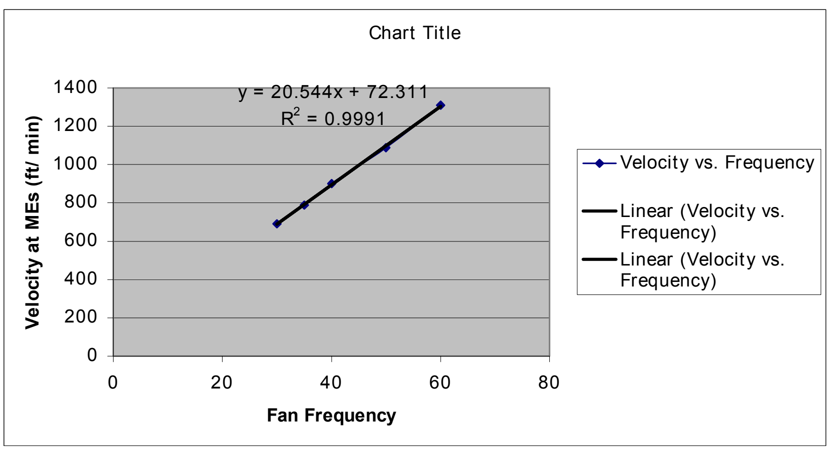
Air Flow Calibration Data

Measurements of Air Flow

18-Jul-01 JJS/ KRW air density 0.0746
and 20 July-2001

					DP data taken on 07-20-01	
Fan Frequency (hz)		**30**	**35**	**40**	**50**	**60**
DP (in H20)						
Meas. Pt.	**1**	0.065	0.08	0.11		
	2	0.07	0.09	0.115		
	3	0.06	0.08	0.11	*(eyeball average by JJS with PJ28@330psig)*	
	4	0.055	0.075	0.09	*0.15-0.16*	*0.22-0.23*
	avg	**0.063**	**0.081**	**0.106**	**0.155**	**0.225**
Velocity below fan (ft/ sec)		16.73	19.08	21.82	26.35	31.75
	fpm	1004	1145	1309	1581	1905
Diam at Fan (in) =	36.5					
Area (ft2)	7.27					
Volumetric Flow		7295	8318	9512	11489	13842
Velocity at DE's		**691**	**788**	**901**	**1088**	**1311**
(Diameter= 44")	44					
Area (ft2)	10.56					

note: droplets breaking away from ME's @ 60 hz



Nozzle Cooling Performance Data

CEC/ EPRI - ACC Performance Enhancement Program
EnviroCare International, Inc. - American Canyon Pilot Tests
Summary of Nozzles Tested

Date		Nozzles Tested	Style (1)	Approx. Flow Rate (gpm) @275 psig
7/ 16/ 2001		PJ -28	P	0.26
		PJ-32	P	0.37
7/ 17/ 2001		24-6	S	0.45
		24-8	S	0.45
		JJS-5	Custom Swirl	
7/ 18/ 2001		24-20	S	
		24-18	S	0.15
		24-8	S	0.45
7/ 25/ 2001		24-3	S	0.45
		20-8	S	0.35
		PJ-15	P	0.3
		24-18	S	0.15
8/ 7/ 2001		PJ-28	P	0.26
		Gianotti	Colliding Jet	0.65
		WDB-24	S	0.5
8/ 8/ 2001		PJ-28	P	0.26
		WD-24	S	0.52
		WD-8	P	0.06
10/ 22/ 2001		PJ-24	P	0.21
		WDB-24	S	0.5
		SSD	Custom Swirl	0.34
(1)	**P**	**Pintle Type**		
	S	**Swirl Type**		

ACC Performance Enhancement Nozzle Test Data Sheet

Date: 07.16.01 **Test Personnel** K. Wilber/ J. Maulbetsch

Testing with Brentwood CDX80 Mist Eliminators

Fan HZ	Air Velocity Ft/ Min	Mist Elim DP(inH2O)	Nozzle	Pressure psig	Flow gpm	Inlet Air WBT	Inlet Air DBT	Exit Air WBT	Exit Air DBT	Dbi-Dbo/ Dbi-Wbi	Comments
40	898	0.08	PJ-28	275	0.26	59.9	73.5	59.9	66.4	52%	**Minimal mist carryover**
40	898	0.08	PJ-28	275	0.26	59.5	73.2	59.5	66.5	49%	
35	793	0.06	PJ-28	275	0.26	59.5	73.0	59.5	65.9	53%	
30	688	0.05	PJ-28	275	0.26	57.7	71.2	57.7	66.5	35%	
35	793	0.06	PJ-32	275	0.375	60.0	72.0	60.0	66.8	43%	Minimal Mist Carryover
35	793	0.06	PJ-32	275	0.375	59.8	72.0	59.8	66.3	47%	
35	793	0.06	PJ-32	275	0.375	59.8	70.5	59.8	63.8	63%	
35	793	0.06	PJ-32	275	0.375	59.8	71.0	59.8	63.3	69%	
40	898	0.08	PJ-32	275	0.375	59.2	71.0	60.0	65.6	46%	Minimal Mist Carryover
40	898	0.08	PJ-32	275	0.375	59.8	72.0	59.8	67.2	39%	
40	898	0.08	PJ-32	275	0.375	59.3	71.0	59.8	65.4	48%	
40	898	0.08	PJ-32	275	0.375	59.7	71.5	59.8	64.7	58%	
30	688	0.05	PJ-32	275	0.375	58.4	72.0	58.4	66.5	40%	Minimal Mist Carryover
30	688	0.05	PJ-32	275	0.375	58.2	72.0	58.2	64.5	54%	
30	688	0.05	PJ-32	275	0.375	58.0	70.5	58.0	63.9	53%	
30	688	0.05	PJ-32	275	0.375	58.0	71.0	58.0	63.2	60%	

Psychrometric data taken with Dry bulb & Relative Humidity Instrument - Considered less accurate than individual dry bulb and wet bulb measurements
Exit Wet Bulb temperature assumed equal to inlet

ACC Performance Enhancement Nozzle Test Data Sheet

Date: 07.17.01 **Test Personnel** **Maulbetsch/ Wilber/ Schwab**

Testing with Brentwood CDX 80 Mist Eliminator

Fan HZ	Air Flow ft/ min	Mist Elim DP(inH2O)	Nozzle	Pressure (psig)	Flow gpm	Inlet Air WBT	DBT	Exit Air WBT	DBT	(Dbi-Dbo)/ (Dbi-Wbi)	Comments
40	898	0.085	24-8	300	0.446	60.5	68	60.5	62.1	79%	Spray water at 91.5 F
40	898	0.085	24-8	300	0.446	60.2	68.3	60.2	60.2	100%	
40	898	0.095	PJ-32	300	0.42	60.2	68.6	60.2	62.3	75%	
40	898	0.095	PJ-32	300	0.42	60.4	68.6	60.4	62.6	73%	
40	898	0.100	PJ-32	300	0.42	60	70.5	60	63.8	64%	
40	898	0.100	PJ-32	300	0.42	60	72	60	66.3	48%	
40	898	0.090	JJS-5	300	0.21	61.1	78.8	68.9	69.0	55%	Custom Swirl Nozzle
40	898	0.095	JJS-5	300	0.21	60.6	78.6	68.6	68.7	55%	

Psychrometric measurements made with Vaisala Dry Bul Humidity Meter.

ACC Performance Enhancement Nozzle Test Data Sheet

Date: 07.18.01 **Test Personnel** Mike Short/ Karl Wilber

Testing with 6 " of Brentwood CF1900 at Mist Eliminator

Fan HZ	Air Flow ft/ min	Mist Elim DP(inH2O)	Nozzle	Pressure	Flow gpm	Inlet Air WBT	Exit Air DBT	Exit Air WBT	DBT	(Dbi-Dbo)/ (Dbi-Wbi)	Comments
30	688	0.05	IP-24	260	0.21	58.8	70	59.3	66.7	29.5%	
40	793	0.06	IP-24	260	0.21	59.4	70.8	59.5	67.7	27.2%	
50	915	0.09	IP-24	260	0.21	58.9	71.3	59.3	67.8	28.2%	Mist Eliminator performs well
30	688	0.05	WD-24	260	0.375	58.3	69.5	59.2	67.0	22.3%	
40	793	0.06	WD-24	260	0.375	58.6	71.6	59.6	68.4	24.6%	
50	915	0.09	WD-24	260	0.375	58.2	70.6	58.2	68.8	14.5%	Mist Elimination still adequate
40	793	0.06	SSD	260	0.335	58.4	70.6	58.4	68.3	18.9%	

ACC Performance Enhancement Nozzle Test Data Sheet

Date: 07.25.01 **Test Personnel** Schwab/ Wilber

Testing with Brentwood CDX 150 Mist Eliminators

Fan HZ	Air Velocity ft/ min	Mist Elim DP(inH2O)	Nozzle	Pressure	Flow gpm	Inlet Air WBT	Exit Air DBT	Exit Air WBT	DBT	(Dbi-Dbo)/ (Dbi-Wbi)	Comments
40	800	0.045	24-3	330	0.45	62.4	70.4	62.3	65.3	63.8%	**Gusty Winds**
40	800	0.045	24-3	330	0.45	62.2	69.8	62.1	65	63.2%	Spray water temp = 90.5
40	800	0.045	24-3	330	0.45	61.8	69.4	62	65.2	55.3%	
40	800	0.045	24-3	330	0.45	61	68.8	64	64.6	53.8%	
50	1100	0.06	WD-20-8	300	0.35	69.4	61.3	61.8	65.0	45.7%	
50	1100	0.06	WD20-8	300	0.35	69.5	61	62	64.7	43.5%	
50	1100	0.09	PJ-15	300	0.3	61.3	68.4	61.6	65.7	38.0%	
40	800	0.06	PJ-15	300	0.3	60.6	68.6	61.2	65.3	41.3%	Winds subsiding
50	1100	0.09	24-18	300	0.15	61.2	69.7	61.8	65.6	48.2%	Precipitous drop in temperature
40	800	0.06	24-18	300	0.15	60.8	69.6	61.2	63.2	72.7%	with drop in fan rpm

ACC Performance Enhancement Nozzle Test Data Sheet

Date: 08.07.01 **Test Personnel** Mike Short/ Karl Wilber

Testing with Brentwood CDX 150 Mist Eliminator

Fan HZ	Air Flow ft/ min	Mist Elim DP(inH2O)	Nozzle	Pressure	Flow gpm	Inlet Air WBT	Exit Air DBT	Exit Air WBT	DBT	(Dbi-Dbo)/ (Dbi-Wbi)	Comments
40	898	0.08	PJ-28	100	0.19	67.5	84.9	68.6	80.9	23.0%	**Minimal mist carryover**
40	898	0.08	PJ-28	100	0.19	67.7	86.1	68.5	80.7	29.3%	
40	898	0.08	PJ-28	300	0.32	67.8	86.6	68.6	79.2	39.4%	More mist carryover
30	688	0.06	PJ-28	300	0.32	67.3	85.9	68.1	75.9	53.8%	
30	688	0.06	PJ-28	300	0.32	67.2	86.8	68.7	76.3	53.6%	
40	898	0.08	PJ-28	300	0.32	68	87.2	68.8	77.9	48.4%	
40	898	0.08	Gianotti	200	0.56	68.2	87.2	69.6	79.5	40.5%	Minimal mist carryove
40	898	0.08	Gianotti	300	0.69	68.4	87.8	70.3	79.8	41.2%	
40	898	0.08	WBD-24	300	0.52	68	86.4	69.5	78.3	44.0%	
30	688	0.06	WDB-24	300	0.52	68	86.2	69.4	76.7	52.2%	

CTI- Type psychrometers used for exit air temperatures.

ACC Performance Enhancement Nozzle Test Data Sheet

Date:)8.08.01 **No Mist Eliminator in Pilot Test Cylinder** **Test Engineer J. S. Maulbetsch, K. R. Wilber**

		Nozzle Pressure (PSI)	Flow (gpm)	Distance from ME (ft)	Fan Hz	Calc. Air Vel. (FPM) at ME	DP Across D.E. (in H20)	Inlet DBT (°F)	Inlet WBT (°F)	Outlet DBT (°F)	Outlet WBT (°F)	?DBT (°F)	(DBTo-DBTi) / (DBTi-WBTi)	Comments
Nozzle	**PJ28**	110	0.18	6.5	30	688	N/A	76.1	64.4	69.8	64.8	6.3	54%	*Droplet Carryover Heavy*
ME	**None**	300	0.32	6.5	40	688	N/A	76.0	64.3	68.1	64.8	7.9	68%	*Droplet Carryover Heavier*
		300	0.32	6.5	40	898	N/A	76.0	64.4	67.6	64.6	8.4	72%	*Droplet Carryover Heavier*
		300	0.32	6.5	40	898	N/A	75.6	64.2	67.6	64.5	8	70%	*Droplet Carryover Heavier*
Nozzle	**WDB24**	300	0.52	6.5	40	898	N/A	75.5	64.0	67.5	64.4	8	70%	*Exit Dry-Bulb & Wet Bulb*
ME	**None**	300	0.52	6.5	40	898	N/A	75.4	64.0	67.4	64.3	8	70%	*housings very wet*
Nozzle	**WDB8**	105	0.035	6.5	40	898	N/A	75.2	64	69.4	64.2	5.8	52%	
ME	**None**	300	0.06	6.5	40	898	N/A	75.0	64.0	68.0	64.0	7	64%	
		350	0.065	6.5	40	898	N/A	75.4	64.2	67.7	64.2	7.7	69%	
		350	0.065	6.5	40	898	N/A	75.3	64.2	67.5	64.1	7.8	70%	

CTI-type psychrometers used for exit air temperatures.

ACC Performance Enhancement Nozzle Test Data Sheet

Date: **10/22/2001** **Test Personnel** Mike Short/Karl Wilber

Testing with 6 " of Brentwood CF1900 at Mist Eliminator

Fan HZ	Air Flow acfm	Mist Elim DP(inH2O)	Nozzle	Pressure psig	Flow gpm	Inlet Air WBT	DBT	Exit Air WBT	DBT	Effectiveness (Dbi-Dbo) / (Dbi-Wbi)	Comments
30	**580**	**0.05**	**IP-24**	**260**	**0.21**	**58.8**	**70**	**59.3**	**66.7**	**29.5%**	**Minimal mist carryover**
40	**760**	**0.06**	**IP-24**	**260**	**0.21**	**59.4**	**70.8**	**59.5**	**67.7**	**27.2%**	**More mist carryover**
50	**915**	**0.09**	**IP-24**	**260**	**0.21**	**58.9**	**71.3**	**59.3**	**67.8**	**28.2%**	
30	**580**	**0.05**	**WD-24**	**260**	**0.375**	**58.3**	**69.5**	**59.2**	**67.0**	**22.3%**	**Minimal mist carryover**
40	**760**	**0.06**	**WD-24**	**260**	**0.375**	**58.6**	**71.6**	**59.6**	**68.4**	**24.6%**	
50	**915**	**0.09**	**WD-24**	**260**	**0.375**	**58.2**	**70.6**	**58.2**	**68.8**	**14.5%**	**Still low carryover**
40	**760**	**0.06**	**SSD**	**260**	**0.335**	**58.4**	**70.6**	**58.4**	**68.3**	**18.9%**	

CTI-Type psychrometers used for exit air dry bulb and wet bulb temperatures

B
SINGLE-CELL TEST DATA

Test data from the single-cell field tests at Crockett Cogeneration Plant are available from the California Energy Commission or the Electric Power Research Institute. The data files include

- Daily test logs
- Tabular output at one-minute intervals and appropriate averages
- Graphs of important relationships

C
COST ESTIMATING FRAMEWORK FOR SPRAY ENHANCEMENT COOLING SYSTEMS

AIR COOLED CONDENSER - INLET AIR COOLING SYSTEM
COST ESTIMATE PROGRAM

PLANT:	**ESTIMATED BY:**
OWNER:	**DATE:**
LOCATION:	
SITE CONTACT:	
PHONE NO.	
email	

Design Conditions			**Design Dry-Bulb**	100 F
	Plant Capacity	**MW**	**Design Wet-Bulb**	70 F
	Steam Flow	**lbs/ br**	**Wet Bulb Depression**	30 F
	Heat Load	**BTU/ HR**	**Desired Inlet Dry Bulb**	80 F
			Dry-Bulb Reduction	20 F

AIR COOLED CONDENSER:
Manufacturer:

No of Cells				15					blue = input data		
Cell Dimensions:				**Length (ft)**	**Width (ft)**						
				35	35						
Tower Dimensions				175	105						
Air Flow/ Cell				1,000,000	**cfm**						
Fan Diameter				30.00	**ft**						
Air Inlet Height				25	ft						
SPRAY SYSTEM:						**Desired Safety Factor**			10%		
						Assigned			**Cost Estimates**		
Spray Water Required				44.3	gpm/ cell	or nominal			**Materials**		**Labor**
Total Spray Water				665	gpm						
Desired Pipe Velocity				7.5	ft/ sec						
Pipe ID Required	Cell			1.6	in		2.0				
	Main Header			5.5	in		6.0				
Spray Pressure				270	psig	No.					
Pump HP Required				119.65	2 pumps	3	100.0	$	31,500		
Pump Skid/ Valve Rack								$	17,948		
Dual Basket Strainer								$	7,500		
SS Water Tank (1000 gal)								$	5,000		
Connecting Piping								$	2,500		
Supply Lines (from Skid to Nozzle Array Pipe Dia (in)				1050							
	Length of	2.0		682.5							
	Length of	6.0		367.5							
Cost per Foot of SS Pipe & Hangers		2.0	$	1.50				$	1,024	$	3,071
Cost per Foot - SS Pipe & Hangers		6.0	$	18.00				$	6,615	$	2,205
No. of Nozzles per Array or Cell				89							

SPRAY SYSTEM (CONT.):							
No of Arrays	15						
Array Headers				$ 22,500		$ 1,500	
Installation					$ 37,500	$ 2,500	per cell
Total Nozzle costs				$ 15,954		$ 12	per nozzle
Controls	***Basic***		1	$ 7,500		$ 7,500	
	PLC			$ -		$ 15,000	
Interface with Control Room					$ 15,000		
Motor Starters				$ 5,000			
Misc. Site Preparation & Labor					$ 10,000		
Equipment Rental (e.g. JLG, etc)				$ 5,000			
System Engineering					$ 10,000		
Engineering Drawings					$ 5,000		
	sq. ft	**$/ft2**					
Windwalls (if Required)	***437.5***	***$ 10.00***		$ 4,375.00			
Installation		***$ 15.00***			$ 6,562.50		
Mist Eliminators	***10603***	***$ 10.00***					
Installation							
	days	$/day					
Construction Management	30	$ 1,000			$ 30,000		
Commissioning & Start Up	15	$ 1,000			$ 15,000		
Training and Operations Manuals					$ 5,000		
	Totals			**$ 132,416**	**$ 139,339**		
	Total Project Est.			**$ 271,755**			